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ASTIA DOCUMENT Nr. AD 142268

EQUIPMENT COOLING SYSTEMS FOR AIRCRAFT

V. H. LARSON

RESEARCH, INCORPORATED

JANUARY 1958

WRIGHT AIR DEVELOPMENT CENTER

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UNITED STATES AIR FORCE
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

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FOREWORD

This report was written by Mr. [illegible] of [illegible], Incorporated, with the assistance of Messrs. [illegible], H.H. [illegible], and L. Irwin. The program was conducted under USAF Contract No. [illegible], Project No. [illegible], "Centralized Vs. Decentralized Cooling Study" and Task No. 6118, under the direction of the Aeronautical Accessories Laboratory, Wright Air Development Center, with Messrs. F.[illegible] Fleischbach and A.L. Tagelfinger acting as WADC project coordinators.

In addition to the authors of this report as noted above, Messrs. A.E. Abramson, [illegible], and [illegible] participated in the project.

The authors express their gratitude to Messrs. Fleischbach and Tagelfinger for numerous helpful suggestions and conferences during the course of the study. Manufacturers of aircraft equipment provided a great deal of valuable technical information. The cooperation of and [illegible] the [illegible] Borg-Warner Corporation and the Stratos Division of Fairchild Engine and Airplane Corporation are gratefully acknowledged.

ABSTRACT

This report contains the results of a study of equipment cooling systems for aircraft operating at flight velocities up to Mach [illegible] and at altitudes up to [illegible] feet. Surface temperatures of the equipment from [illegible]° to [illegible]°F are considered.

Systems are analyzed to determine the effect of flight velocity and altitude on the weight, drag, and power requirement of the cooling system. The relative effect of the different equipment cooling systems on aircraft performance for various flight conditions is considered by translating power and drag to an equivalent weight that would have approximately the same effect on the aircraft performance.

Vapor cycle cooling systems using Freon-11 and systems using water are analyzed to determine the weight of various components and the power requirement for the systems. Simple and regenerative air cycle systems and a ram air cooling system combined with an expendable coolant system are also analyzed. Expendable coolant systems used singly and in combination with other cooling systems are considered for high speed dash flight of limited duration. The range of applicability and characteristics of the various systems are analyzed. The applicability of each of the systems to centralized and to individualized concepts of equipment cooling is considered.

PUBLICATION REVIEW

The publication of this report does not constitute approval by the Air Force of the findings or conclusions contained herein. It is published only for the exchange and stimulation of ideas.

FOR THE COMMANDER:

[illegible]
Chief, Air Conditioning [illegible]
[illegible] Section
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INTRODUCTION

Modern aircraft are equipped with numerous pieces of equipment that must be cooled during flight and during ground operation. The need for cooling of aircraft electrical, electronic, and mechanical equipment arises from the fact that such equipment will operate only over a limited range of temperature, that operation above some maximum temperature is impossible or inefficient, and that all such equipment dissipates heat. The dissipated heat must be removed if steady-state operation is to be within the inherent temperature limitations and must be transferred to a heat sink which is external to the equipment.

The removal of the heat dissipated by equipment in aircraft operating at the relatively low velocity and low altitudes typical of a few years ago involved simply air circulation by means of a blower or free convection. As flight speeds and altitudes increased, more involved cooling systems such as air cycle systems became necessary. The problem has become progressively more acute with the present trend toward much higher velocities and altitudes. For example, aerodynamic heating results in a total air temperature of approximately 500°F at Mach 2.5. At the atmospheric pressures which prevail at an altitude of 70,000 feet, the heat transfer coefficient (for air in turbulent flow) is about one-tenth as large as at sea level with equal velocity and temperature. This factor makes it much more difficult to remove heat even if air at a low temperature is available, e.g., from an air cycle cooling system. The problem is further aggravated by the fact that as flight speeds increase, many more functions are performed by electronic equipment thus greatly increasing the actual cooling loads. An analysis of cooling systems for flight speeds up to Mach 1.8 is presented in reference 1. The present study is primarily concerned with flight speeds from Mach 1 to Mach 2.5 and altitudes from sea level to 70,000 feet.

Aircraft equipment can be cooled by transferring the heat to a sink at a temperature below the equipment temperature or by "pumping" the heat to a sink at a higher temperature. Direct air, air cycle systems, and expendable coolant systems are examples of the first type. Vapor cycle systems utilize the heat pump principle.

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Equipment design offers a means of alleviating the problem. Design which reduces the heat dissipation rates is one approach to the problem. Another approach is equipment designed to operate at higher temperatures. Both factors should be considered because the weight and power required to remove heat varies directly with quantity of heat and inversely with the temperature at which it must be removed. Consequently, reduction of heat dissipation rates may be more than offset if the temperature level must be reduced. Of course, the cost of producing the power that must be dissipated must be included when determining temperature and heat dissipation effects. The equipment design objective should be high temperature operation with low heat dissipation rates, i.e., equipment that will operate efficiently at a high temperature.

In this study, various types of systems are analyzed. Vapor cycle systems pump the heat from the equipment temperature to the higher atmospheric recovery or total temperature where the heat is transferred to the atmosphere. The basic components of such systems are the evaporator, compressor and condenser, and a refrigerant together with lines and any auxiliary components needed to transport the heat to or from the system. Air cycle systems utilize heat exchangers and a turbine to cool engine bleed air (or simply a turbine to cool ram air) so as to have the sink at a temperature below the equipment temperature. Ducts to conduct the air and a compressor to increase the air pressure and to load the turbine are needed for a complete cooling system. Systems may use expendable coolants which undergo a change of state as heat is absorbed from the equipment. The coolant is then exhausted overboard in a higher energy state. Such systems require an evaporator, storage tank, lines and, in some cases, auxiliary heat transport provision.

Cooling systems transferring the heat to the fuel as a heat sink are also considered for some applications.

In addition to the four basic types of systems - vapor cycle, air cycle (and simple air cooling) expendable coolant, and fuel sink systems - numerous modifications of each system and combinations of the various systems are analyzed.

The cooling system has certain characteristics which must be considered in the design of a particular system or in comparison of different possible designs. Some of the factors, such as weight power consumption, and drag, have a direct effect on aircraft performance. Others may have an indirect effect on performance or may have an effect that makes use of a system prohibitive or undesirable. Among such factors are space requirement, vulnerability, safety, convenience, and cost. In an actual case, the various factors must all be weighed so as to determine the most desirable cooling system. In general, the first three factors, weight, power, and drag having a direct effect on aircraft performance, are most readily evaluated. The other factors can then be used to select between systems that are approximately equal on the basis of performance or as justification for a system that may have somewhat greater detrimental effect considering only the first factors.

In this study, the effect of the various factors affecting the performance of aircraft are considered on the basis of range, rate of climb, and flight duration.

The various cooling systems have been evaluated on the basis of what will be called an engineering optimization. The effects of the significant design variables have been determined in so far as possible, and optimum or near optimum values have been assigned. In some instances, an actual mathematical optimization has been made, in other cases graphical optimization was deemed more feasible, and in still other cases intuitive reasoning has been considered adequate. A rigorous mathematical optimization of a cooling system is out of the question because of the number of variables involved and because there is no common basis for (nor effect of) the various factors. Factors other than those having a direct effect on aircraft performance may frequently be decisive but do not enter into the mathematical expressions, in which case a mathematical optimization would be irrelevant. Further, the mathematical complexity of a formal optimization analysis would frequently obscure rather than clarify the cooling system evaluation and thus defeat the purpose of the study.

An effort has been made to assign typical values to any factors involving somewhat arbitrary assumptions or pertaining to flight schedules, aircraft or engine characteristics not specifically known. In all cases, the assumptions that have been made are clearly stated and the effect of different assumptions is pointed out.

SECTION I

ATMOSPHERIC AND FLIGHT CONDITIONS

The range of altitude and of flight velocity being considered in this study is shown in figure 1. Lines of constant true airspeed are included in the figure. The temperature range versus altitude is shown in figure 2. The maximum temperature that may be encountered at a particular altitude will impose the most stringent requirement on a cooling system. Consequently, all calculations in this study are made assuming the ambient temperature is at the maximum indicated by the temperature-altitude envelope of figure 2.

The total ram temperature at a particular flight condition is equal to the ambient temperature plus the kinetic temperature rise and is given by the equation

$$T_T = T_a + \left(\frac{\gamma - 1}{2}\right) M^2 \tag{1}$$

The total temperatures (calculated for the maximum ambient temperatures) are plotted on an altitude-Mach number envelope in figure 3. The range of total temperatures considered in this study thus varies from about 50°F at Mach 1 and 65,000 feet altitude to 525°F at Mach 2.5 and 35,000 feet altitude.

The recovery temperature or adiabatic wall temperature will be of interest for any system utilizing a surface-type heat exchanger. The recovery temperature is

$$T_r = T_a + r\left(\frac{\gamma - 1}{2}\right) M^2 \tag{2}$$

Values of the recovery temperature assuming a recovery factor (r) of 0.85 are plotted on the altitude-Mach number envelope in figure 5. The recovery temperatures which apply for this study vary from about 40°F at Mach 1 and 65,000 feet altitude to 450°F at Mach 2.5 and 35,000 feet altitude.

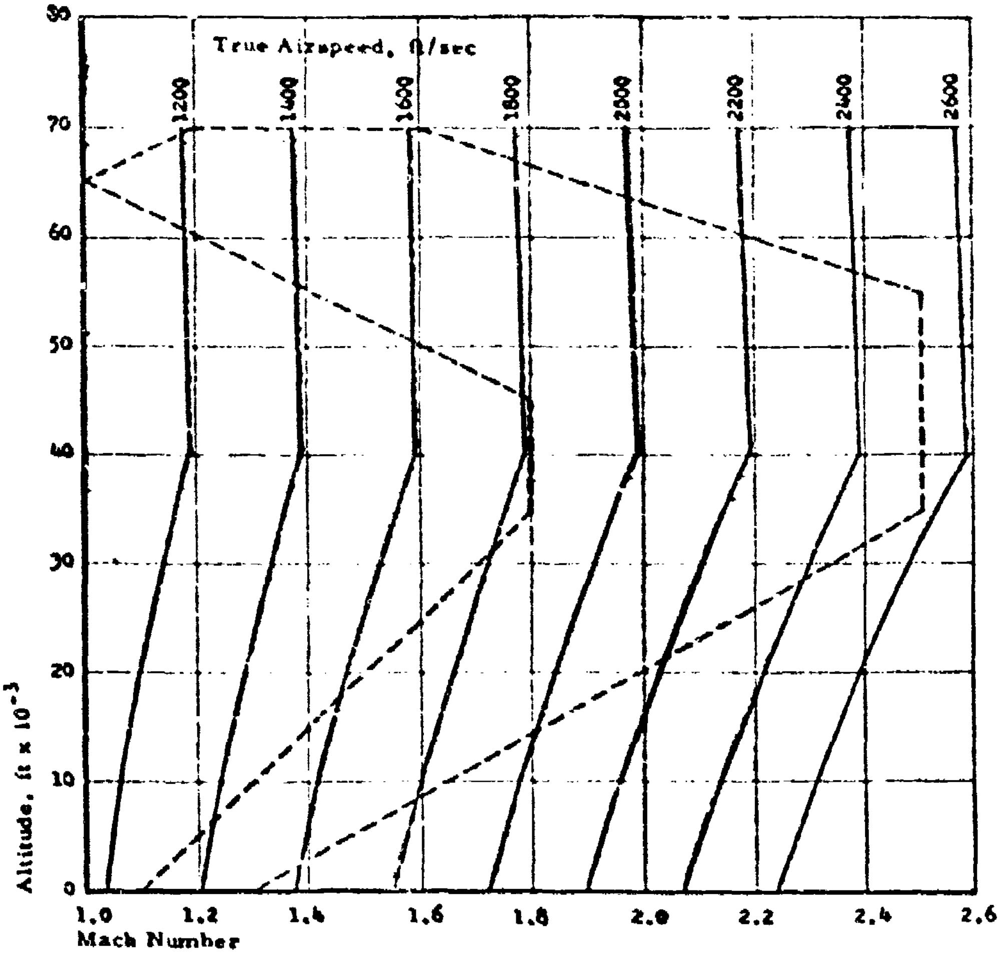


FIGURE 1 FLIGHT ALTITUDE VERSUS MACH NUMBER INCLUDING TRUE AIRSPEED CURVE

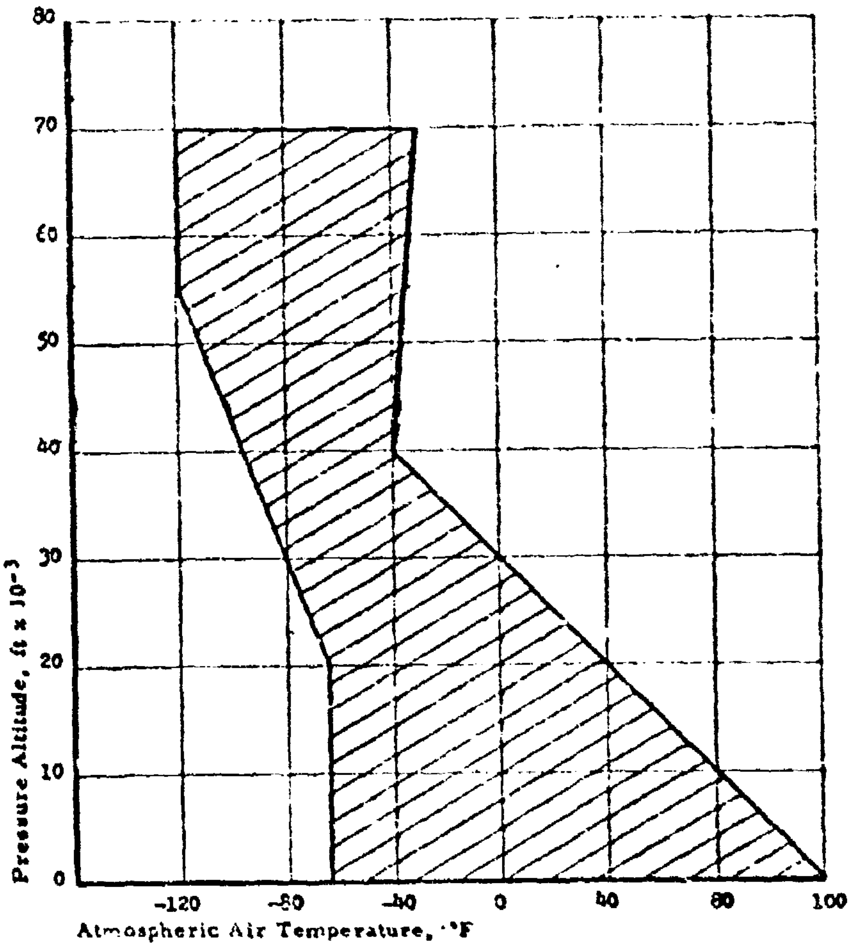


FIGURE 2 VARIATION OF AMBIENT TEMPERATURE WITH ALTITUDE

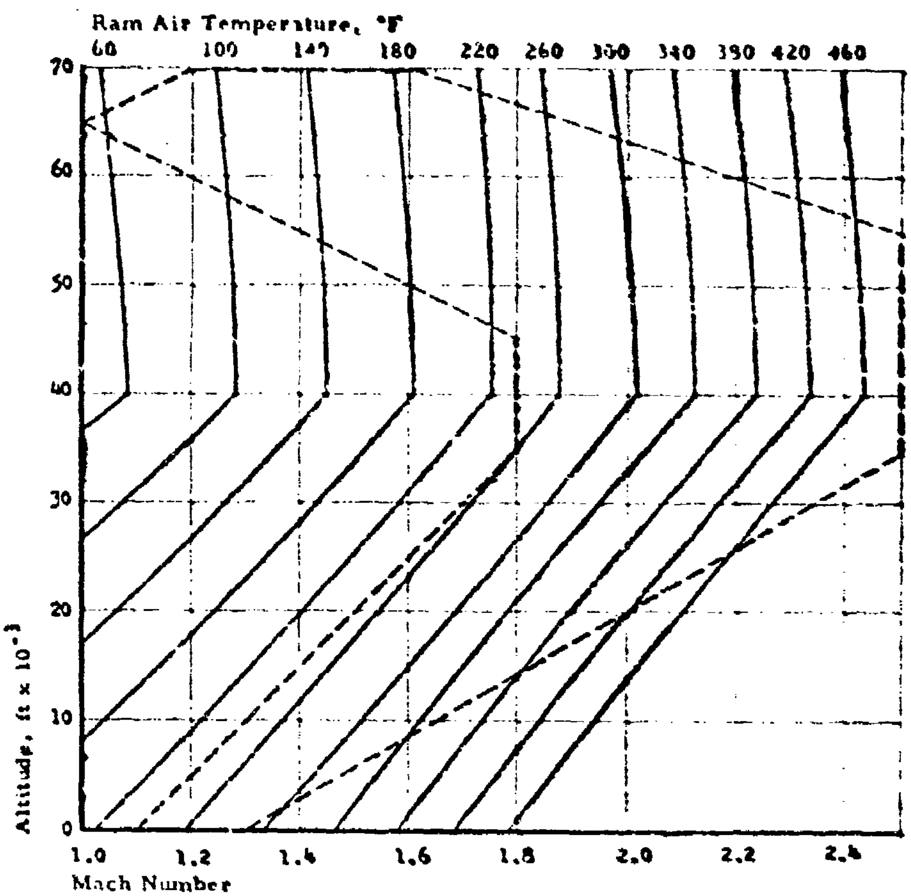


FIGURE 3 VARIATION OF RAM AIR TEMPERATURE OVER SPECIFIED FLIGHT ALTITUDE AND MACH NUMBER RANGE

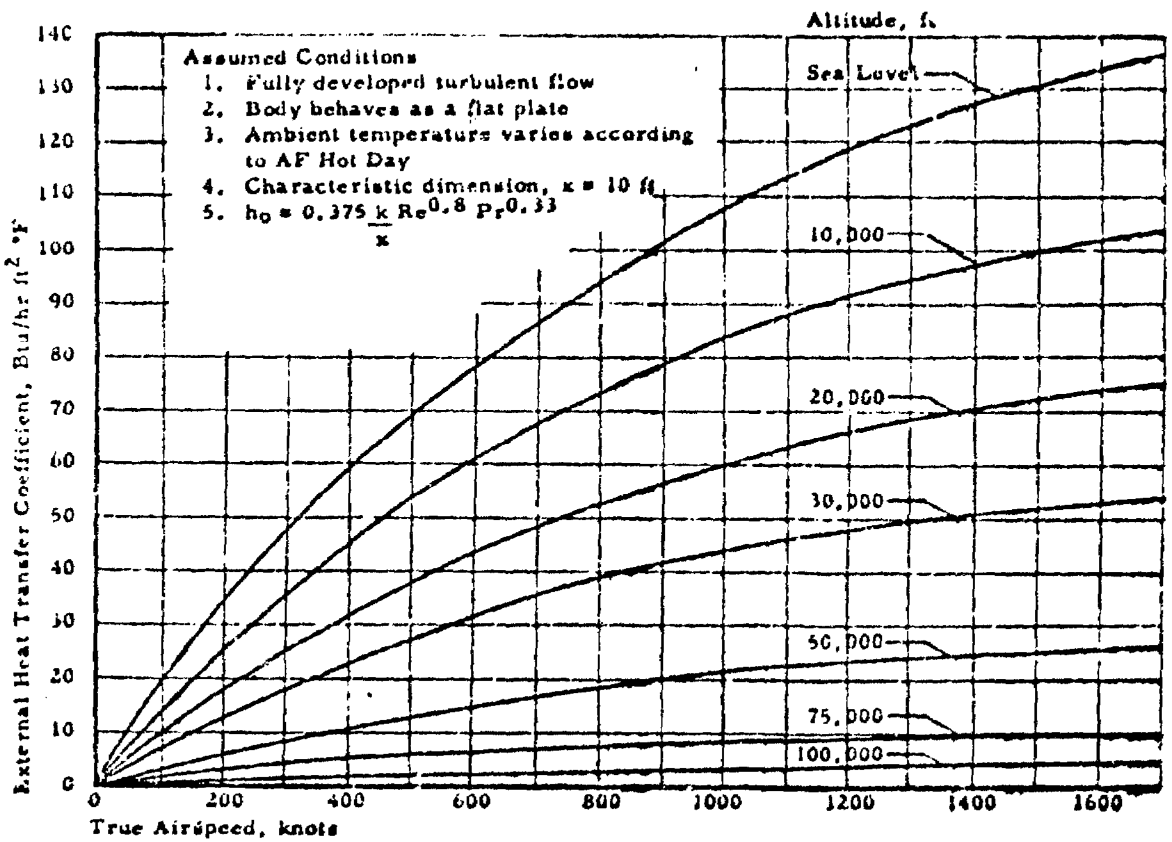


FIGURE 4 EXTERNAL HEAT TRANSFER COEFFICIENT VERSUS VELOCITY

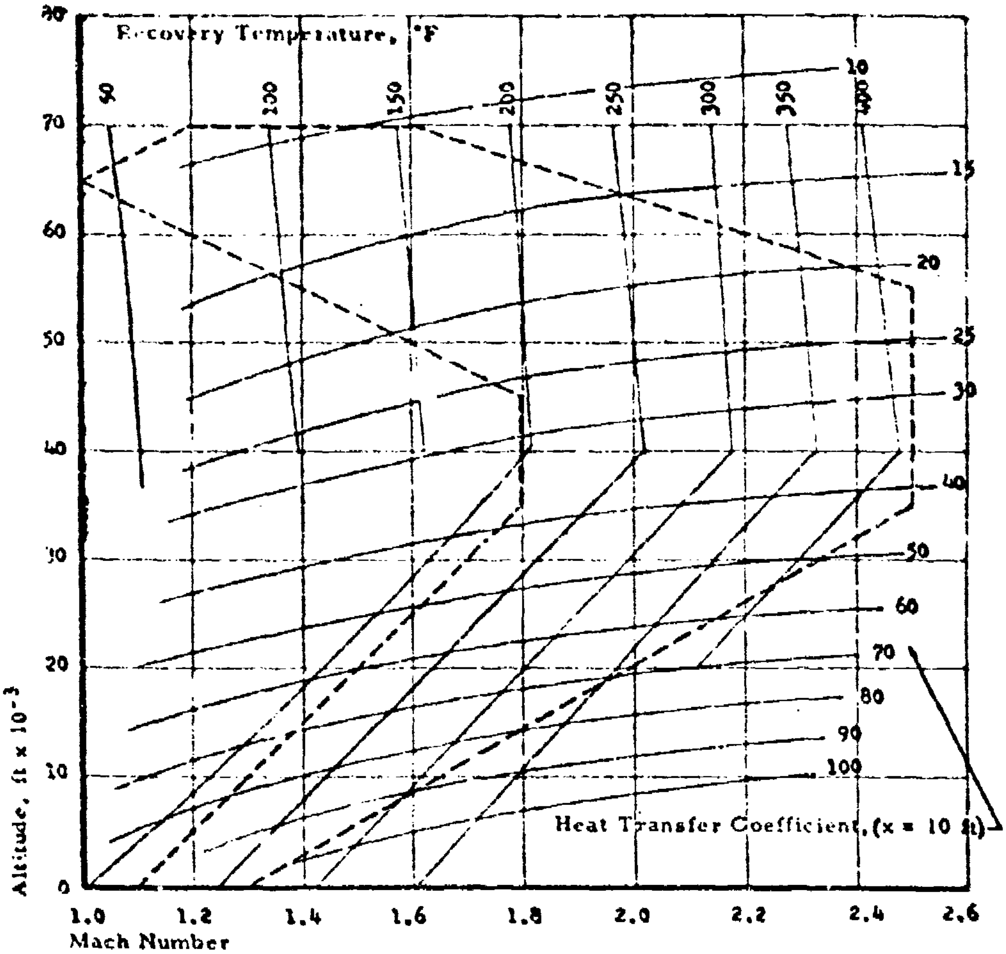


FIGURE 5 VARIATIONS OF RECOVERY TEMPERATURE AND CONVECTIVE SURFACE HEAT TRANSFER COEFFICIENT WITH ALTITUDE AND MACH NUMBER

For any system utilizing surface-type heat exchangers, the heat transfer coefficients that apply on the surface are of particular interest. The heat transfer coefficient at a distance x, assuming uniform temperature, is given by the equation

$$Nu_x = 0.0375 \, (Re)_x^{0.8} \, (Pr)^{0.33} \qquad (3)$$

The heat transfer coefficients that apply for the hot day and for the Mach number and altitude range of interest in this study are plotted in figures 4 and 5.

The data for figures 4 and 5 were taken from figure 4 of reference 2. The original data were plotted according to the method of reference 3. The Prandtl number is averaged over the applicable temperature range to give a value of $(Pr)^{0.33} = 0.89$. The other properties of air are evaluated at an intermediate temperature $T' = T_a (1 + 0.133 M^2)$ as given in the reference. The Reynolds number was calculated assuming that the viscosity of the air varies directly as the 0.7 power of the absolute temperature, T', so that

$$Re = (5.88 \times 10^6) \, (p_a \, T_a^{0.5} \, Mx) \qquad (4)$$

The heat transfer coefficients that apply for the altitude-Mach number envelope of interest in this study are replotted in figure 5 as lines of constant heat transfer coefficients on the altitude-Mach number envelope. Lines of constant recovery temperature, assuming a recovery factor of 0.85, are also plotted in figure 5.

The minimum temperature considered in this study is -65°F; consequently, any system must be designed so as not to be damaged by exposure at that temperature (e.g., damage by freezing). This requirement has not be interpreted as excluding use of any fluids that may freeze at temperatures above -65°F, providing the system could be designed to provide continuous cooling without the use of auxiliary means of thawing out the cooling system. In other words, any system utilizing a fluid that freezes at temperatures above -65°F must be designed so that the heat dissipated by the equipment being cooled will thaw out any portions of the system that may freeze so that the system is capable of continuous equipment cooling from -65°F to the maximum design condition.

Continuous flight is assumed except for systems utilizing an expendable coolant in which case systems are evaluated on the basis of total flight time for a simple expendable coolant system or for the time of a high speed dash for the cruise-dash-cruise flight, in which case a combination system is assumed. In the latter case, the system would be capable of continuous cooling at the design cruise condition.

The effect of changes in the assumed atmospheric and flight conditions can be quite accurately predicted by referring to figures 3 and 5. The trends of total and recovery temperature increase with the square of the Mach number. The surface heat transfer coefficients decrease quite rapidly as the altitude increases above 60,000 feet. The heat transfer rates that can be secured by circulating air will also be greatly reduced at the higher altitudes, thus greatly increasing the weight of any heat exchangers that must transfer heat to the air at the lower air pressures.

SECTION II

EQUIPMENT CONDITIONS

The equipment to be cooled is not specifically defined for this study. The equipment is simply a "box" that dissipates heat at a rate from 200 watts to 1 kw for each item. It is assumed that the temperature limitations of the equipment will not be exceeded if a surface is provided at the equipment at a specified temperature. Surface temperatures of 160° to 275°F are assumed. The actual temperature of equipment elements will then depend on the effective overall heat transfer per unit temperature difference between the element and the surface.

Systems utilizing heat transfer media that exhibit a great variation in wc_p cannot be directly compared because of the differences that exist at the equipment component. A system in which the heat is absorbed by air will have a wide variation in air temperature as the air picks up the heat. This is a direct result of low wc_p values for airflow. Typically, the air temperature variation would be in the order of 100° to 200°F. Systems utilizing a liquid heat transfer media for which the values of wc_p are usually many times greater than for air can be designed so that the fluid temperature variation is in the order of 10°F or even less. Systems utilizing a fluid that changes state at the equipment have essentially a constant fluid temperature at the equipment.

A comparison of systems with such widely different characteristics is therefore somewhat arbitrary and will in general depend on the requirements of a particular application. Since the lowest temperature that can be obtained in the equipment is above the inlet temperature, systems utilizing air in the equipment will be capable of cooling some elements to a much lower temperature than systems using a liquid transfer fluid. This is particularly desirable for equipment which is comprised of elements that have different temperature requirements. For this case, efficient cooling is possible by arranging the elements so that those requiring low temperatures are near the inlet and the higher temperature elements near the fluid exit point.

The heat dissipated by the equipment must be transferred to a fluid or sink at a lower temperature level. Except for the case where the sink has infinite thermal capacity, or where the heat transfer results in a change in state of the fluid, the fluid temperature will vary as it is circulated through the equipment. The change in temperature of a liquid or gas within the equipment is

$$T_{Ee} - T_{Ei} = \frac{Q_E}{3600\ Wc_p} \tag{5}$$

The temperature at which the fluid leaves the equipment, T_{Ee}, is a measure of the minimum temperature difference that will exist in the equipment component, that is, T_{Ee} is the highest equipment fluid temperature. The equipment component or some elements within the component must be at a higher temperature than the fluid exit temperature. In this sense, the exit temperature, T_{Ee}, is a measure of the cooling potential of the cooling system. As such, T_{Ee} is perhaps as valid as a comparison parameter for most cases as other means of comparing systems with widely varying characteristics. Using T_{Ee} as a comparison is then simply assuming that different systems are comparable when T_{Ee} is equal and can then be evaluated on the basis of weight and power. Other alternatives are a defined effective equipment surface temperature or an average equipment fluid temperature.

An effective equipment surface temperature for heat transfer that is dependent on the temperature of the fluid entering the equipment, and the fluid exit temperature, can be defined in terms of an equipment effectiveness analogous to the usual heat exchanger effectiveness,

$$e_E = \frac{T_{Ee} - T_{Ei}}{T_{ES} - T_{Ei}} \tag{6}$$

Then the effective equipment surface temperature is

$$T_{ES} = T_{Ei} + \frac{T_{Ee} - T_{Ei}}{e_E} \tag{7}$$

This temperature is an average equipment surface temperature for heat transfer defined by the assumed average effectiveness.

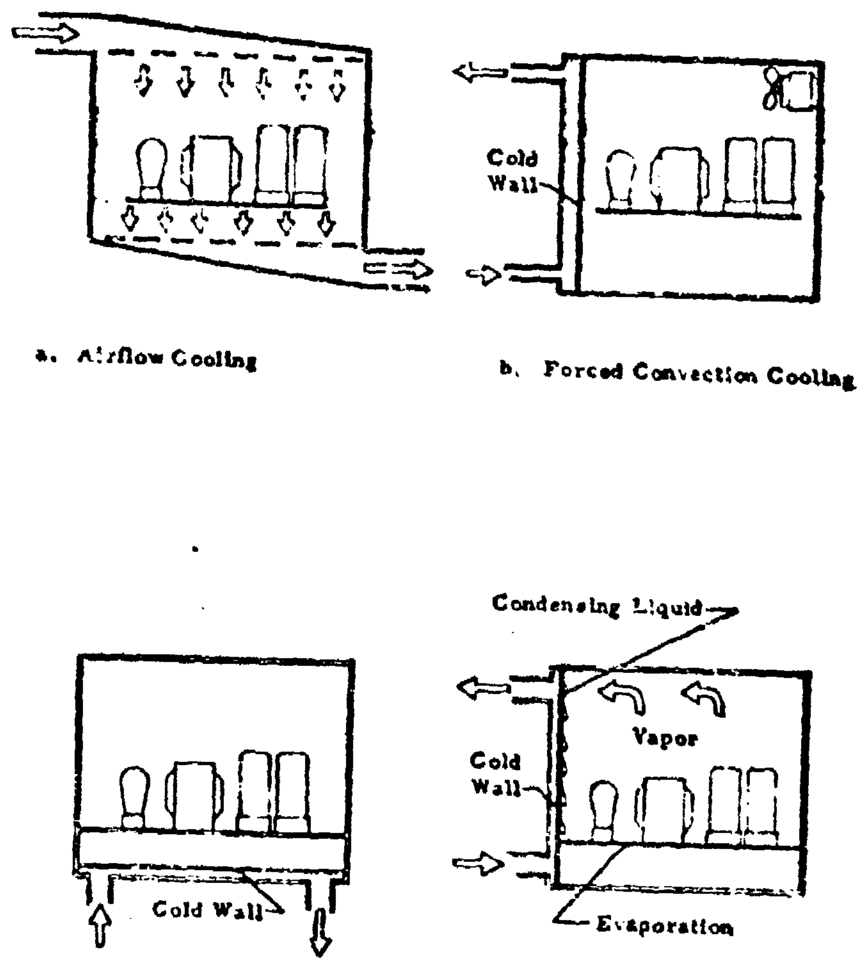
The average temperature of the fluid in the equipment may also be of interest for a comparison of different types of systems which may operate with very small fluid temperature change (e.g., a vapor cycle system) or with a relatively large change in fluid temperature (e.g., an air cycle system). An average equipment temperature may be applicable for comparison of cooling systems provided that the equipment contains elements that can operate at various temperatures so that elements requiring a low temperature can be effectively cooled by the entering fluid, while elements capable of higher temperature operation can be cooled by the hotter fluid near the fluid exit point.

The means of cooling at the equipment is not specified in detail. The actual heat transfer means will depend on the type of cooling system. The system may simply circulate air over the equipment as illustrated schematically in figure 6a. This cooling means is particularly adapted to air cycle systems. A liquid heat transport fluid may be used. In this case, the heat must be transferred by convection, conduction, radiation, or by means of an evaporating and condensing fluid. Systems utilizing a liquid transport fluid, or an evaporator at the equipment, will be considered as cold wall systems, the heat being picked up by the cold wall and then transferred to a heat sink or to a heat pump. Examples of such systems are shown schematically in figure 6b, -c, and -d.

As pointed out above, a comparison of systems with conditions at the equipment as different as in the case for the air coolant system (figure 6a) and the other systems (figure 6b, -c, and -d) is rather arbitrary. For this study, a comparison on the basis of equal equipment fluid exit temperature T_{E_e} has been selected as the most realistic. It is then assumed that the actual equipment component is properly designed to be efficiently cooled by the airflow or by the cold plate.

The maximum distance from any piece of equipment to the cooling system is specified as 150 feet. The assumed distribution for this study is that the major portion of the cooling load is relatively near (about 20 feet) with a portion of the load at up to the 150-foot maximum distance. All systems are analyzed assuming the 20-foot distance, the dispersed load is then analyzed to determine its effect on weight and power requirement.

The effect of different assumptions can be determined by noting the trends and relative effects of the equipment conditions as pointed out in the section of this report on applicability and evaluation. In general, the systems show a marked increase in weight and power as the equipment temperature decreases. The effect of dispersed equipment is for most cases relatively less significant. The weight and power are approximately directly proportional to the cooling load for given temperature and cooling conditions.



a. Airflow Cooling

b. Forced Convection Cooling

c. Conductive Cooling

d. Evaporative Cooling

FIGURE 6 MEANS OF TRANSFERRING HEAT AT THE EQUIPMENT

SECTION III

EVALUATION CRITERIA

The objective of a cooling system evaluation is to determine the relative merits of the various systems, to define the range of applicability, and to point out certain features that may be significant depending on the type of aircraft and its mission.

In the final analysis, the cooling system should be analyzed along with the aircraft in general and with the equipment being cooled in particular. For this study, it is assumed that the cooling load and equipment temperatures have been specified. All equipment items should be designed for an optimum between minimum heat dissipation and maximum operating temperature as well as for efficient transfer of heat to the cooling fluid.

The fundamental factors which should be considered in an evaluation of a cooling system are

1) Weight
2) Power requirement
3) Drag
4) Space requirement
5) Reliability
6) Vulnerability
7) Safety
8) Ground operation
9) Convenience
10) Cost

Some of the above factors affect aircraft performance; others may affect aircraft design and dependability. In this study, primary evaluation is based on the factors which affect performance; other factors are relegated to a secondary role with a sort of "veto" power. Such factors are specifically considered and discussed for those cases in which widely varying characteristics are noted.

The evaluation of an equipment cooling system requires the determination of the magnitude of a number of factors which have an adverse effect on the performance of an aircraft as compared with the performance of the aircraft assuming operation is possible without the cooling system. The factors directly affecting aircraft performance are weight, power requirement, and drag. Usually, since it can be assumed that the cooling system is an essential piece of equipment, a comparison of various systems based on their effect on aircraft performance is of as much significance as is the determination of specific effects on aircraft performance. Such a comparison, being more general, is frequently of greater value that a detailed analysis of an effect. A comparison of this type results in an equation in which some of the aircraft performance parameters will cancel, resulting in an evaluation criteria that is quite general and can be applied to many different cases even without detailed performance information.

In general, the cooling system exhibiting the smallest values of the factors adversely affecting performance is preferable. In the actual case, the relative magnitude of weight and power, for example, will vary widely for different types of systems and, further, it is frequently possible to reduce the power required by increasing weight. It is therefore necessary to have some means of translating the various factors of weight, power requirement, and drag to some common unit. For this study, an equivalent weight has been selected as the common unit because all systems and all components possess weight. The actual weight can then be used directly. It is then necessary to determine the power input to the system and any drags imposed by the system and to translate these factors into the equivalent weight unit.

The power (or drag) equivalent weight is then that weight that would have approximately the same effect on the aspect of aircraft performance being considered as a given power requirement (or drag imposition). The total equivalent weight of a system is the sum of the actual weight, the power equivalent weight, and the drag equivalent weight.

The factor used to translate power and drag to an equivalent weight will depend on the type of aircraft, its flight characteristics, and on the aspect of performance chosen as the most significant in a particular case.

Among the aspects of performance which can be used as evaluation criteria are range, flight duration, and rate of climb. The first two would appear particularly significant for bomber and transport aircraft, while the rate of climb may be more significant for fighter-type aircraft.

The approximate effect of a small change in the various factors can be determined by writing an equation describing the aspect of performance selected as a criteria and taking the partial derivative of that equation with respect to the factors affected by the cooling system. The change in performance is approximately equal to the change in the factor times the partial derivative of the performance equation with respect to the factor being considered. This method of determining the effects of the various factors is based on the following assumptions:

1) The change or increment is small compared with the basic factor.

2) The flight pattern is essentially the same with the changed factors.

3) The equation describing the aspect of performance is sufficiently accurate with and without the changes imposed by the cooling system.

4) Any secondary effects can be neglected.

This method of evaluation requires an equation describing the aspect of performance in terms of the factors affected by the cooling system. A number of such equations are given in the literature. References 4 and 5 give the Breguet range and endurance equations and a range equation for jet aircraft.

An evaluation criteria based on the range of an aircraft can be defined by using the Breguet range equation

$$Rg = (C\,\eta/SFC)(L/D)\ \ln\left[W_T/(W_T - W_f)\right] \qquad (8)$$

The partial derivative with respect to total weight is

$$\frac{\delta Rg}{\delta W_T} = -(C\,\eta/SFC)(L/D)\left[W_f/(W_T - W_f)\,W_T\right] \qquad (9)$$

The fractional change in range for a small change in weight is then

$$\frac{\Delta R_g}{R_g} \doteq \ln\left[\frac{(W_T - W_f)}{W_T}\right]\left(\frac{W_f}{W_T - W_f}\right)\left(\frac{\Delta W}{W_T}\right) \tag{10}$$

The specific fuel consumption can be written in terms of average power (P), initial fuel weight (W_f), and time that the fuel will last (t).

$$SFC = W_f/(P\,t) \tag{11}$$

The partial derivative of equation (8) with respect to power is

$$\frac{\partial R_g}{\partial P} = \frac{C\,\eta\,t}{W_f}(L/D) \ln\left[\frac{W_T}{W_T - W_f}\right] \tag{12}$$

The fractional change in range caused by the small change in power is then

$$(\Delta R_g/R_g) \doteq (\Delta P/P) \tag{13}$$

The partial derivative of equation (8) with respect to drag is

$$\frac{\partial R_g}{\partial D} = \frac{C\,\eta}{SFC}\,\frac{L}{D^2} \ln\left[\frac{W_T}{W_T - W_f}\right] \tag{14}$$

The change in range caused by a small change in drag is then

$$(\Delta R_g/R_g) \doteq -(\Delta D/D) \tag{15}$$

The effect of power and drag can be expressed in terms of an equivalent weight unit that would have approximately the same effect on the range of the aircraft by equating the fractional change in range as given by equations (10) and (15).

$$\frac{\Delta P}{P} \doteq \ln\left[\frac{W_T - W_f}{W_T}\right]\left(\frac{W_f}{W_T - W_f}\right)\left(\frac{\Delta W}{W_T}\right) \tag{16}$$

Then

$$\left.\frac{\Delta W}{W_T}\right|_{P\,eq} \doteq -\left(\frac{W_T - W_f}{W_f}\right) \ln\left[\frac{W_T}{W_T - W_f}\right]\left(\frac{\Delta P}{P}\right) \tag{17}$$

Similarly, the drag equivalent weight is

$$\left(\frac{\Delta W}{W_T}\right)_{D\ eq} \doteq \left[\left(\frac{W_T - W_f}{W_f}\right) \ln \left(\frac{W_T}{W_T - W_f}\right)\right]\frac{\Delta D}{D} \tag{18}$$

The total equivalent weight, the sum of the actual weight, the power equivalent weight, and the drag equivalent weight can be expressed in pounds by multiplying through by the gross weight,

$$\Delta W_{T\ eq} \doteq \Delta W + \left[\left(\frac{W_T - W_f}{W_f}\right) \ln \left(\frac{W_T}{W_T - W_f}\right)\right]\left(\frac{W_T}{D}\Delta D - \frac{W_T}{P}\Delta P\right) \tag{19}$$

Letting $X = W_f/W_T$, the initial fuel weight to gross weight ratio, and assuming a constant lift-drag ratio, equation (19) can be written

$$\Delta W_{T\ eq} \doteq \Delta W + \left[\left(\frac{1 - X}{X}\right) \ln \left(\frac{1}{1 - X}\right)\right]\left(\frac{L}{D}\Delta D - \frac{550\ L}{DV}\Delta IP\right) \tag{20}$$

An evaluation criteria based on the range equation for jet aircraft can be derived in an analogous manner. The result is similar to the Breguet range criteria except for the factor indicating the effect of the fuel-to-gross-weight ratio.

The range of a jet aircraft can be expressed by the equation (reference 4)

$$Rg = \frac{C}{SFC}\ \frac{L}{D}\left[1 - \left(1 - \frac{W_f}{W_T}\right)^{0.5}\right] \tag{21}$$

The partial derivative of equation (21) with respect to a gross weight is

$$\frac{\partial Rg}{\partial W_T} = -\frac{1}{2}\frac{C}{SFC}\ \frac{L}{D}\left(1 - \frac{W_f}{W_T}\right)^{-1/2}\ \frac{W_f}{W_T^2} \tag{22}$$

The total equivalent weight determined in a manner entirely analogous to that for the Breguet criteria is

$$\Delta W_{T\ eq} = \Delta W + \frac{2\left[X - 1 + (1 - X)^{1/2}\right]}{X}\left(\frac{L}{D}\Delta D - \frac{W}{P}\Delta P\right) \tag{23}$$

The flight duration of an aircraft flying a Breguet schedule can be expressed by the equation

$$Du = \frac{750}{SFC}\ \frac{L}{DV}\left[\frac{1}{(1-X)^{1/2}} - 1\right] \qquad (24)$$

The partial derivative with respect to gross weight is

$$\frac{\partial Du}{\partial W_T} = \frac{750}{SFC}\ \frac{L}{DV}\left(\frac{-1}{2}\right)\frac{X}{(1-X)^{3/2}}\ \frac{1}{W_T} \qquad (25)$$

The total equivalent weight based on flight duration for a Breguet schedule is then

$$\Delta W_{T\ eq} = \Delta W + \frac{2\left[1 - X - (1-X)^{3/2}\right]}{X}\left(\frac{L}{D}\Delta D - \frac{W}{P}\Delta P\right) \qquad (26)$$

The three evaluation factors are thus functions of the initial fuel-to-gross-weight ratio, of the lift-drag, and weight-power ratios. The three functions of the fuel-to-gross-weight ratio are plotted on a common axis in figure 7.

For fighter and interceptor type aircraft, the rate of climb may be a more significant performance factor than flight range or duration. A criteria based on rate of climb can be derived in a manner similar to that used for the range criteria. The rate of climb of an aircraft is given by the equation

$$C = (dH/dt) = (TV - DV)/W \qquad (27)$$

The partial derivative of equation (27) with respect to weight is

$$(\partial C/\partial W) = -(TV - DV)/W^2 \qquad (28)$$

The fractional effect of a small change in weight on the rate of climb is

$$(\Delta C/C) \doteq -(\Delta W/W) \qquad (29)$$

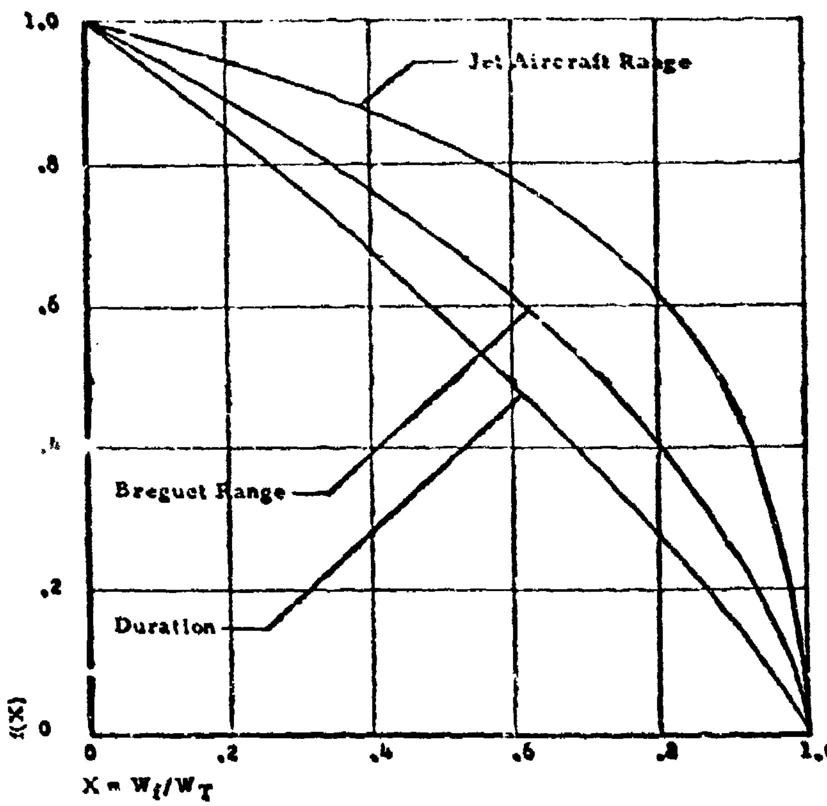


FIGURE 7 FACTORS FOR CONVERTING POWER AND DRAG TO AN EQUIVALENT WEIGHT BASIS

The partial derivative of equation (27) with respect to power (TV) is

$$\frac{\partial C}{\partial (TV)} = \frac{1}{W} \qquad (30)$$

The fractional change in the rate of climb caused by a small change in power is

$$\frac{\Delta C}{C} \doteq \frac{\Delta (TV)}{CW} \qquad (31)$$

The partial derivative of the rate of climb with respect to drag is

$$(\partial C/\partial D) = -(V/W) \qquad (32)$$

The fractional change in the rate of climb caused by a small change in drag is

$$(\Delta C/C) \doteq -\frac{V}{CW} \quad \Delta D \qquad (33)$$

The total equivalent weight for the rate of climb criteria (with the power expressed in horsepower) is

$$(\Delta W)_{T\ eq} \doteq \Delta W + \frac{V\Delta D}{C} - \frac{550\Delta HP}{C} \qquad (34)$$

The curves of figure 7 indicate that with the stated assumptions the calculated effect of a power requirement or drag imposition relative to the effect of a weight increase depends on the initial fuel-to-gross-weight ratio, on the equation assumed to define the aircraft performance in the most significant manner, and on the lift-drag or weight-power ratio.

In this study, typical values are assigned to the translation factor for expressing drag and horsepower requirements in terms of an equivalent weight, e.g., the drag translation factor ($f_{D\ eq}$) in the case of Breguet range is $(L/D)\left[(1 - X)/X\right] \ln \left[1/(1 - X)\right]$. The total equivalent weights are calculated on the basis of the assigned values. The approximate effect of using translation factors other than the assigned values can be readily determined by noting the effect of the different assigned values. The actual horsepower, drag, and actual weights are indicated for a number of cases, thus permitting comparison of these basic effects.

In the systems evaluation for this study, the total equivalent weights are calculated using power translation factors ($f_{P\ eq}$) with assigned values of one and two. The drag translation factors ($f_{D\ eq}$) were assigned values of two and three. As pointed out above, the actual value of such translation factors depends on the L/D ratio for the aircraft and on the fuel-to-gross-weight ratio. In general, the L/D ratio decreases with increasing flight velocity; therefore, cooling system evaluation on the basis of small translation factors is representative for high speed flight.

It should be noted that since the values assigned for this study differ by one, for each case, the difference in the total equivalent weights for otherwise similar conditions is numerically equal to the drag or to the horsepower required.

SECTION IV

HEAT TRANSPORT FLUIDS

The heat dissipated by the equipment will not, in general, be in the immediate vicinity of the cooling system and therefore the heat must be transported to the cooling system or alternately the heat sink must be brought to the equipment before being dispelled overboard. In this report, the phrase "heat transport" is used to describe the process whereby heat is moved between relatively distant points, as distinguished from heat transfer which will be reserved for heat flow between a fluid and a surface. Vapor cycle, regenerative air cycle, and some of the expendable coolant systems would most likely utilize some heat transport means. Air cycle systems with ducts to conduct the air to the equipment and some types of expendable coolant systems are examples of systems in which the sink is brought to the equipment and then expelled overboard. Such systems do not involve heat transport in the sense used here.

The transport of heat consists of a simple fluid flow circuit with the fluid undergoing cyclic temperature changes, i.e., absorbing heat at the equipment and giving off the heat at the cooling system. The heat transport portion of a cooling system is comprised of the lines, a circulating pump, and the heat transport fluid. It is assumed that the fluid does not undergo any change of state. Fluids that change phase in a system are considered in section VII of this study, vapor cycle cooling systems.

The heat transport is achieved by virtue of a temperature change in the fluid, an increase in temperature at the equipment, and a decrease in temperature at the cooling system. In addition to the heat absorbed in the equipment, the heat added by the fluid pump due to friction in the pump and lines must be removed. This latter heat addition is very small and will therefore be neglected in this analysis.

The amount of heat that is absorbed by a transport fluid is

$$Q = 3600w\ c_p\,(T_{E_e} - T_{E_i}) \qquad (55)$$

For a given cooling load, the temperature rise will vary inversely with the product of the mass flow rate and the specific heat of the fluid (wc_p). Since weight is one of the major factors in a cooling system, it is advantageous to have a fluid with a high specific heat. A dense fluid is desirable so as to secure a high mass flow rate with small tubes and low flow velocity.

The effects of the transfer fluid on aircraft performance are caused by fluid weight, the weight of the lines and passages at the equipment and at the cooling system that contain the heat transport fluid, and the weight of the pump, electric motor and associated electrical equipment. The power required to circulate the fluid will also have an effect on aircraft performance.

Among the more important properties for a heat transport fluid are the following:

1) Freezing point must be below minimum temperature encountered so fluid can be pumped.

2) Vapor pressure should be relatively low at the applicable temperatures so that excessive pressures are not necessary to prevent formation of vapor. Vaporization cannot be tolerated because of the resultant low heat transfer rates and the possibility of vapor lock.

3) Thermal conductivity should be high so as to secure high heat transfer coefficients.

4) Specific heat should be high, reducing the required mass flow rate.

5) Density should be high so as to reduce volume flow and size of lines.

6) Viscosity should be low to reduce pumping power.

In addition to the above qualities, such factors as toxicity, corrosive tendencies, availability, and cost should be considered. The properties of particular interest in this study of several of the fluids that appear most promising as heat transfer fluids up to a temperature of 275°F are listed in table 1 (reference 6).

TABLE 1 PROPERTIES OF HEAT TRANSPORT FLUIDS

	Conductivity (Btu/hr ft °F) 200 °F	Viscosity (Centipoises) 200 °F	Specific Heat (Btu/lb °F) 200°F	Specific Weight (lbs/ft^3) 200°F	Vapor Pressure (psia) 200°F	Vapor Pressure (psia) 250 °F
Water	0.393	0.305	1.00	60.1	11.5	29.8
Water-Ethyl Alcohol Solution	0.140	0.455	0.84	50.4	23.8	
Water-Methyl Alcohol Solution	0.193	0.36	0.677	5[illegible].5	29.0	68.9
Water-Ethylene Glycol Solution	0.209	0.97	0.830	63.8	7.3	21.3

An analysis of the various systems utilizing heat transport fluids indicates that the effect of differences among the better transport fluids, on the total transport system weight, is rather small and also that the transport system is usually a small part of the total equivalent weight of the system. In this study, a fluid transport system with ethylene glycol as the fluid has been assumed for all cases up to a temperature of 275°F. The use of any of the other fluids listed in table 1 will not appreciably alter the total equivalent weights. The ethylene glycol solution was chosen primarily because of its relatively low vapor pressure and for safety considerations, it being a non-toxic and non-combustible fluid.

Heat transport at the high condenser temperatures requires a fluid with special properties, particularly a fluid that is stable and has a low vapor pressure at the applicable temperature. Properties of several fluids at 500°F are listed in table 2 (references 6, 7, and 8). In this study, Dowtherm is assumed to be the transport fluid for all systems requiring heat transport at temperatures above 300°F. The selection was made primarily on the basis of vapor pressure.

The heat dissipated by the equipment must be transferred through a tube or passage wall and then to the heat transport fluid. The means of transferring the heat to the wall from the equipment is discussed in section II of this report. The transfer of the heat to the fluid is by forced convection. The required area is a function of the temperature differences, the heat transfer coefficient, and the cooling load. Since it is very desirable to minimize temperature drops and area, it is advantageous to secure high heat transfer coefficients.

The significance of the heat transport fluid on the overall cooling system performance and on its total equivalent weight varies widely depending on the type of system considered. The weight flow rate times the specific heat (wc_p) for the heat transport fluid determines the difference between the equipment inlet and exit temperature. The equipment exit temperature is specified, consequently the inlet temperature is determined by the heat transport system. The cooling system must provide a temperature below the equipment inlet temperature. Any system for which the weight or power is sensitive to the minimum temperatures that must be attained is affected much more by the heat transport system than is a system in which low temperatures are more readily secured.

TABLE 2 PROPERTIES OF HIGH TEMPERATURE HEAT TRANSPORT FLUIDS AT 500 F

	Conductivity (Btu/hr ft °F)	Viscosity (Centiposses)	Specific Heat (Btu/lb °F)	Specific Weight (lbs/ft^3)	Vapor Pressure (psia)
Water	0.356	0.11	1.13	49.0	680.8
Dowtherm A	0.105	0.35	0.63	53.1	15.65
Dowtherm E	0.123	0.20	0.565	65.2	60.6
Minnesota Mining Fluorochemical 0-75	0.02 to 0.010	0.15	0.31	81.7	10.1
Minnesota Mining Fluorochemical N-43	0.03 to 0.01	0.12	0.32	80.5	2.0
Dow Corning Silicon Fluid DC-550	0.076	3.0	0.65	54.9	low
Ucon Fluid 50-HB-280-X	0.092	1.8	0.63	53	0.29

Vapor cycle systems, because of the significance of evaporator temperature (the minimum temperature in the system) on the compressor size and weight and on the power requirement, are relatively sensitive to the heat transport system. The regenerative air cycle, on the other hand, because of the low temperature characteristic of the system, is relatively unaffected by the heat transport system.

The equivalent weight of the heat transport system can be approximated by the empirical equation

$$W_{eq} = (4 + 2kw) \frac{\ell}{100} \tag{36}$$

The value of ℓ should be the line length or twice the distance from equipment to the cooling system. Equation (36) is a good approximation provided the transport system equivalent weight, as given by the equation, does not drop below about six pounds. In that event, a reasonable estimate of weight is accurate enough for comparison of various systems.

Equation (36) is an empirical approximation to the equivalent weight of a heat transport system in which the tube size and fluid flow velocity have been optimized when used as a heat transport system with a vapor cycle cooling system. The optimization is with respect to a minimum total equivalent weight for the cooling system and transport system. While somewhat lighter heat transport systems could be used with some systems, the regenerative air cycle, for example, the change is not considered significant. Consequently, equation (36) may be used for nearly all cases.

As pointed out above, the use of suitable transport fluids other than the water-ethylene glycol solution assumed in this study will have a relatively minor effect on the total equivalent weights used for cooling system evaluation.

SECTION V

COMPRESSION VAPOR CYCLE COOLING SYSTEMS

One of the methods for cooling aircraft equipment at high flight speeds is by means of a compression vapor cycle cooling system. This method transfers heat from the equipment by utilizing external energy to pump the heat to a sink at a higher thermal potential. The heat pumping fluid undergoes a change of state in going through the various parts of the cycle. The cooling effect is obtained because of the heat required to change the state of the fluid from the liquid to the vapor state.

A. Basic Considerations

Vapor cycle cooling systems are shown schematically in figures 8 and 9. In these systems, heat is transported from the equipment to the evaporator by means of a circulating heat transport fluid. The heat is then absorbed at a relatively low temperature level by evaporation of the refrigerant.

The major components of a vapor cycle cooling system include

1) The refrigerant or working fluid

2) The evaporator in which the heat is absorbed

3) A compressor to circulate the refrigerant and to increase the pressure and temperature

4) A condenser to reject the heat to the heat sink (normally the atmosphere

5) An expansion valve to control the refrigerant flow and reduce the pressure.

A compressor power supply system, the necessary controls, and in some cases separate components for heat absorption at the equipment and for heat transport complete the typical vapor cycle cooling system.

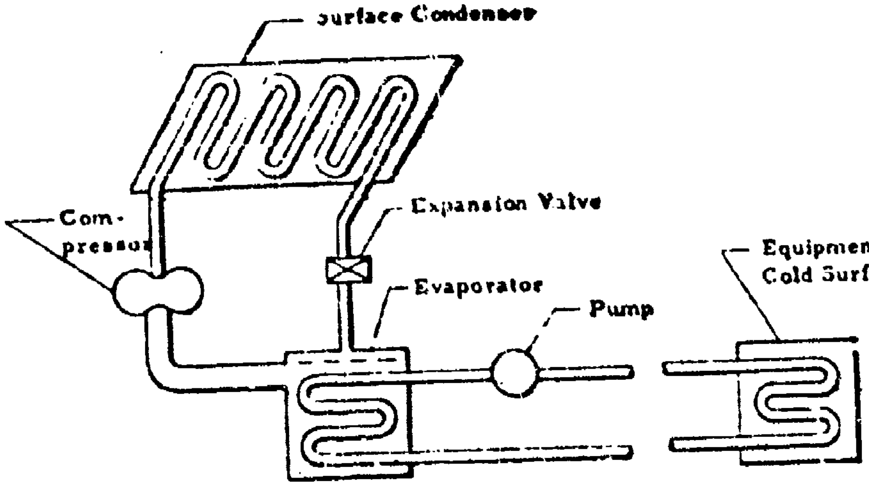


FIGURE 8 VAPOR CYCLE COOLING SYSTEM WITH SURFACE CONDENSER

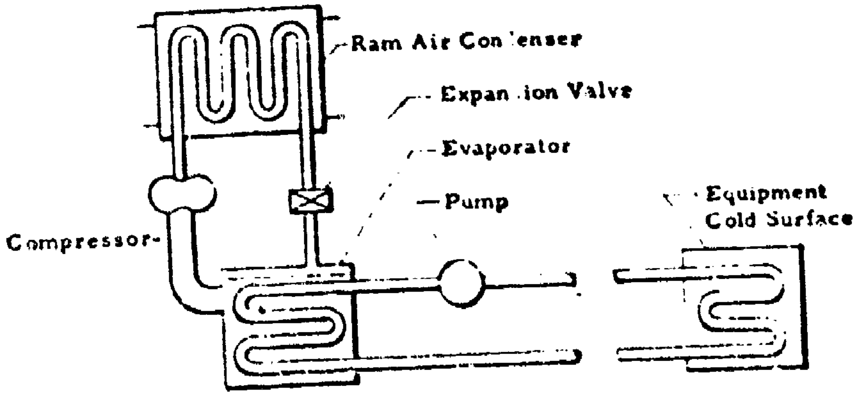


FIGURE 9 VAPOR CYCLE COOLING SYSTEM WITH RAM AIR CONDENSER

The liquid refrigerant is introduced (by means of a pressure-reducing expansion valve) into an evaporator maintained at a pressure such that the boiling point of the liquid refrigerant is at the desired temperature. After absorbing heat in the evaporator, the fluid in a higher energy level vapor state is transferred to a condenser in which the pressure is such that the vapor will liquefy when heat is removed. The heat of vaporization is thus absorbed by the fluid at the lower evaporator temperature and given up to a heat sink at the higher condenser temperature. Energy must be supplied to bring the vapor from the evaporator pressure and temperature to the higher condenser pressure and temperature.

The fluid which is used in a vapor cycle system undergoes cyclic changes in state and temperature and, by virtue of such changes, is used to transfer heat from one temperature to a higher temperature. The following refrigerant property values are of prime interest as they will affect the range of applicability and the performance of the system:

1) Critical temperature

2) Freezing point

3) Latent heat of vaporization

4) Specific heat of liquid

5) Specific heat of gas at constant pressure

6) Specific heat of gas at constant volume

7) Vapor pressure

The critical temperature defines the maximum condenser temperature. The minimum evaporator temperature is determined by the freezing point. These two values do not directly enter into the equation defining cooling effect or power input but are factors of prime importance. The fluid must be above the freezing point throughout the cycle, thus being in the liquid or vapor state, so it can be readily circulated through the system. The critical temperature is that point above which the vapor cannot be liquefied regardless of pressure and, as the cycle requires condensation at the sink temperature, the critical temperature defines an upper limit for the condenser temperature. In practice, the maximum temperature must be considerably below the critical as the refrigerating effect decreases and the work of compression increases rapidly as the temperature approaches the critical value.

The other property values listed above have a direct effect on the power requirement and performance of the cycle. A detailed study of the effect of these factors, as well as the effect of temperature levels, follows the mathematical derivation of the performance characteristics of the cycle. Briefly, considering each factor independently, the desirable qualities in a refrigerant are as follows:

1) High latent heat to secure high cooling effect per pound of refrigerant

2) Low specific heat of the liquid to secure greater cooling effect

3) Low specific heat of the vapor at constant pressure for small work of compression

4) High specific heat of the vapor at constant volume for small work of compression

5) Vapor pressure near atmospheric pressure at evaporator temperature so the cycle will operate at low pressure

6) Small variation of vapor pressure with temperature so as to work at low compression ratio with resultant small work of compression

In actual fluids, the factors are interdependent so that some of the above desirable qualities are mutually exclusive, e.g., high latent heat will usually accompany high specific heats. Consequently, a balance of qualities must be determined, considering the magnitude of each effect and the range of values for actual fluids.

B. Analysis of the Compression Vapor Cooling Cycle

In this preliminary analysis, pressure losses are neglected and efficiencies of 100% are assumed. The effect of actual pressure losses and typical efficiencies are included in the detailed analysis of components and in weight determination.

1. Cooling Effect at the Evaporator

The cooling effect of the vapor cycle per pound of refrigerant is dependent on the latent heat of vaporization, the specific heat of the liquid, and the temperature difference between the evaporator and condenser. Part of the heat of vaporization is supplied by cooling of the liquid refrigerant from the condenser temperature to the evaporator temperature, the balance is supplied in the evaporator and is the net cooling effect of the cycle. The cooling effect, assuming constant specific heat, is given by the equation

$$Q_V = L_V - c_{p_f} \, (T_K - T_V) \quad \text{Btu/lb refrigerant} \qquad (37)$$

The heat of vaporization, L, decreases as the temperature increases and is zero at the critical temperature. Assuming constant specific heats, the heat of vaporization, L_V, at temperature T_V is related to that at a temperature T by the equation

$$L_V = L - (c_{p_f} - c_{p_g}) (T_V - T) \qquad (38)$$

Equations (37) and (38) indicate that there is a definite limit to the temperature difference and to the condenser temperature for which a particular fluid can be used as a refrigerant. The critical temperature, because of the variation of specific heats of actual fluids, is usually less than would be indicated by equation (38). The practical limit of the condenser temperature depends on the critical temperature of the fluid and the allowable power input.

2. Work of Compression

Operation of the vapor cycle requires the transfer of the refrigerant in the vapor state from the evaporator temperature and pressure to the higher temperature and pressure that prevail in the condenser. The necessary energy is added to the vapor by means of a mechanical compressor. The work of compression (per pound of refrigerant) is dependent on the pressure ratio, the initial pressure and volume, and on the compression exponent.

The work of compression in general is least for isothermal compression, in which case the exponent (n) in the equation $pv^n = c$ is equal to one; however, this criteria would not apply to compression for a cooling system because the vapor temperature must be increased to the condenser temperature, thus precluding constant temperature compression. The ideal case for the vapor cycle would be an exponent n such that the temperature after compression is just equal to the condenser temperature, the case of no superheating during compression. In practice, there will be a varying n, $n > \gamma$ (where γ is the isentropic compression exponent) for the first part of compression and $n < \gamma$ for the balance of the compression because of heat transfer considerations. For fluids with a pronounced tendency to superheat during compression and with an efficiently cooled compressor, the effective value of n may be somewhat less than γ and the temperature rise less than the adiabatic temperature rise. For such cases, an analytical treatment, assuming $n = \gamma$, will be conservative as this assumption would indicate a temperature after compression greater than the actual temperature. For fluids that have little or no tendency to superheat during compression or for compressors not effectively cooled, the assumption $n = \gamma$ would be nonconservative.

If compression is assumed to follow a polytropic curve, i.e., $pv^n = c$, the work of compression (Btu/lb) is given by the equation

$$Q = \frac{n}{n-1}\frac{p_1 v_1}{J}\left[\left(\frac{p_2}{p_1}\right)^{(n-1)/n} - 1\right] \qquad (39)$$

In case the compression is isentropic ($pv^\gamma = c$), the work of compression can be expressed by substituting for n

$$Q = \frac{\gamma}{\gamma-1}\frac{p_1 v_1}{J}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad (40)$$

where $\gamma = c_p/c_v$

$p_1 v_1 = RT_1$

$R = J(c_p - c_v)$

Substituting these values, the work of isentropic compression is

$$Q = c_{pg} T_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad (41)$$

In an actual compression system, the compression ratio will be greater than the ratio of condenser to evaporator pressure because of line and valve pressure losses between the cylinder (or other compressive element) and the evaporator, and also between the compressor cylinder and the condenser. Allowance must be made for such pressure drops to determine actual performance; however, in the initial phases of comparative studies, such factors can be neglected. The effect of pressure drops is considered on a power input basis and is disucssed in a subsequent section of this report.

3. Heat Dissipated in the Condenser

The heat generated by virtue of compression, in addition to the heat absorbed by the evaporator, must be dissipated by the condenser. That heat, assuming isentropic compression and neglecting efficiency effects is given by the sum of equations (37) and (41).

$$Q = L_V - c_{p\ell} \; (T_K - T_V) + c_{pg} \, T_V \left[\left(\frac{p_K}{p_V} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \qquad (42)$$

or, using condenser and maximum compression temperature,

$$Q = L_K + c_{pg} \, (T_{max} - T_K) \qquad (43)$$

4. Pressure Reduction

An expansion valve is provided to reduce the pressure from that in the condenser to the evaporator pressure. The expansion is a constant enthalpy process. Consequently, no heat is transferred.

5. Power Input

Vapor refrigeration cycles are usually evaluated on the basis of the coefficient of performance, which is defined as the refrigerating effect divided by the power required to drive the compressor. For this analysis, the power input (per unit of cooling), which is the inverse of the coefficient of performance, will more clearly indicate the effect of changes in condenser temperature, in evaporator temperature, and in property values. The power input as used here is the energy that is added to the refrigerant and therefore does not reflect compressor efficiency or pressure drop.

The power input (PI) for a vapor cycle cooling system, assuming isentropic compression and constant specific heats, is given by the dimensionless equation

$$PI = \frac{c_{pg}\, T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]}{L_1 - c_{p\ell}(T_2 - T_1)} \tag{44}$$

Equation (43) can be readily solved for any condition for which the vapor pressures are known, assuming constant specific heats, or, by integrating the Clapeyron equation, the pressure ratio can be eliminated from equation (43) giving an expression for an ideal vapor cycle, PI, that is dependent only on temperatures and the refrigerant properties. An analysis of coefficient of performance with similar assumptions is presented in reference 9.

For the "ideal" vapor, it is assumed that the liquid volume is negligible, the specific heats are constant, and the vapor obeys the [illegible].

The Clapeyron equation can then be written

$$dp/dT = J\, L/T\, v \tag{45}$$

Eliminating v by the perfect gas relation,

$$pv = R\, T$$

equation (45) can be written

$$\frac{dp}{p} = \frac{JLdT}{RT^2}$$

The heat of vaporization at temperature T is

$$L = L_1 + (c_{pg} - c_{p\ell})(T - T_1) \tag{46}$$

Substituting this value in Clapeyron equation, gives

$$\frac{dp}{p} = \frac{J}{R}\left[\frac{L_1 - (c_{pg} - c_{p\ell})\, T_1}{T^2} + \frac{c_{pg} - c_{p\ell}}{T} \right] \tag{47}$$

Int

$$\frac{p_2}{p_1} \quad \left(\frac{T_2}{T_1}\right)^{\left(\frac{c_{pg}}{c_{pg} - c_{vg}}\right)} \qquad (48)$$

B

$$\left(\frac{p_2}{p_1}\right) \quad \Big)^{(1 - c_{pg}/c_{pg})} \qquad (49)$$

Sub

$$PI \quad \Big)^{\left(\frac{c_{pg}}{c_{vg}} - 1\right)} \qquad (50)$$

r ...uirement given as a fu ... combinations.

L

$$(3-1) \qquad (51)$$

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Equation (51) gives the ideal vapor cycle power requirement and is particularly valuable to illustrate the effect of temperatures and of property values, and as an indication of practical operating ranges for refrigerants of various properties. It must be remembered that this equation, being for an idealized vapor, does not take the critical temperature or the freezing point into account and can therefore be applied to a specific fluid only when the maximum temperature is well below the critical. The actual cycle will require somewhat more power than the idealized cycle of equation (51) and that power input will increase without limit as the condenser temperature approaches the critical value.

6. Effect of Refrigerant Properties

The ideal vapor cycle power input (PI) versus condenser temperature for a range of property values and for an evaporator temperature of 160°F is shown in figure 10. The variation of PI with condenser temperature for several of the Freons considered as idealized vapors with constant properties is shown in figure 11. The upper ends of the curves are inaccurate as the temperature is then approaching the critical where the actual power input becomes infinite. The Carnot cycle, an ideal theoretical cycle, dependent only on temperatures, is shown on each graph. Because of thermodynamic considerations, no cycle (actual or theoretical) can have a lower PI than the Carnot cycle. The ideal vapor cycle power input can be approximately determined for any refrigerant operating between T_1 and T_2 if the property values are known. The curves indicate that, at the lower temperatures considered and for the smaller temperature differences, the usual range of properties do not have a very marked effect. For a temperature difference of 100°F, for example, the maximum difference in PI is about 10% for the range of values considered. (The properties of the common refrigerants are included in this range.) As the temperature difference and the condenser temperature increase, the variation of PI with A is very marked. Refrigerants with a low A are therefore eliminated for the higher temperature applications. The PI decreases as B decreases, but this factor has a relatively much smaller effect than does A, particularly at the higher temperatures. For actual fluids, the value of B is always greater than one and for many refrigerants is approximately 1.5. For high temperature refrigeration, the significance of A, therefore, completely overshadows B and is a factor of prime significance in determining the range of applicability of a particular fluid.

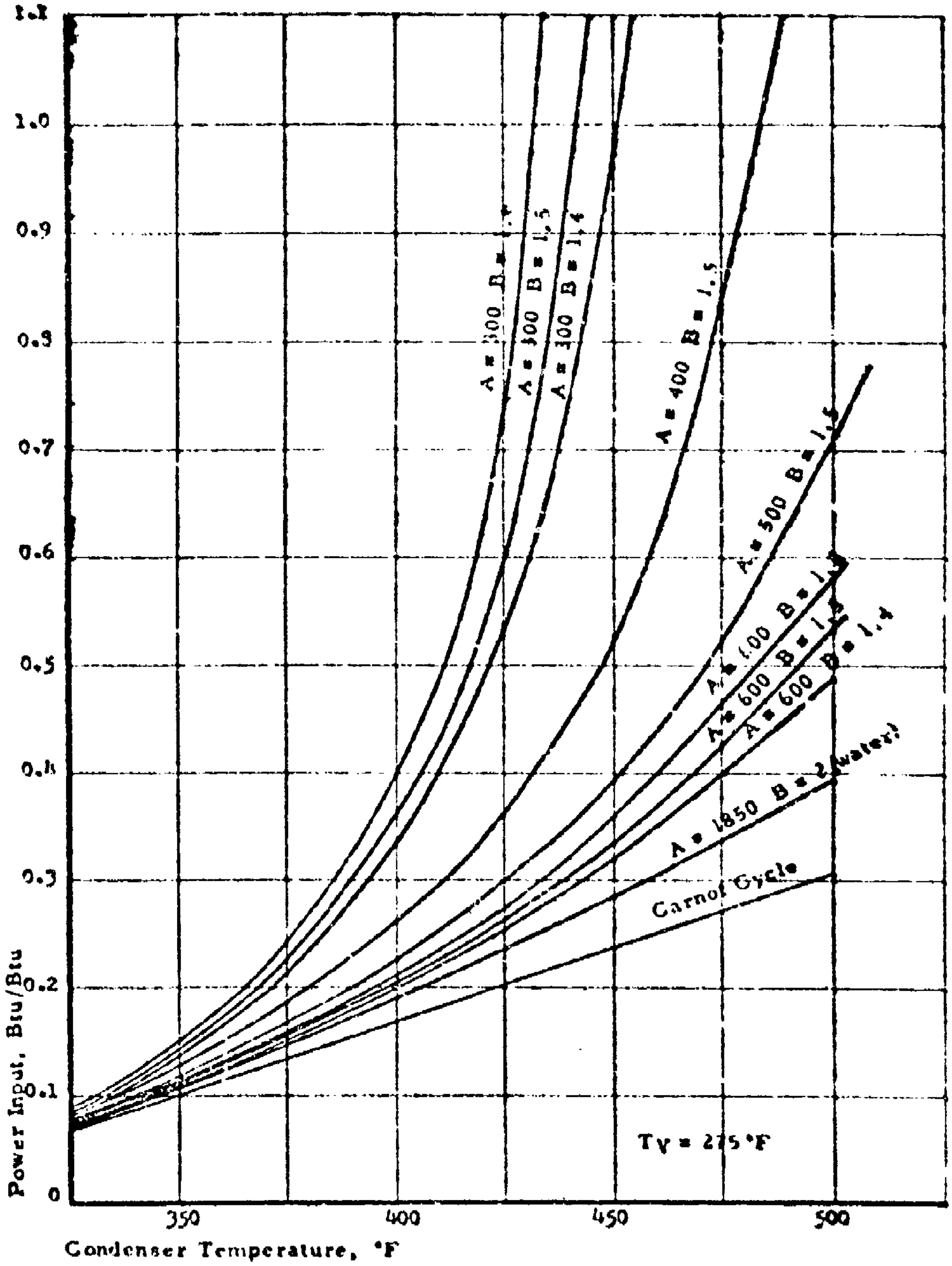


FIGURE 10 EFFECT OF REFRIGERANT PROPERTIES ON THE IDEAL VAPOR CYCLE POWER INPUT

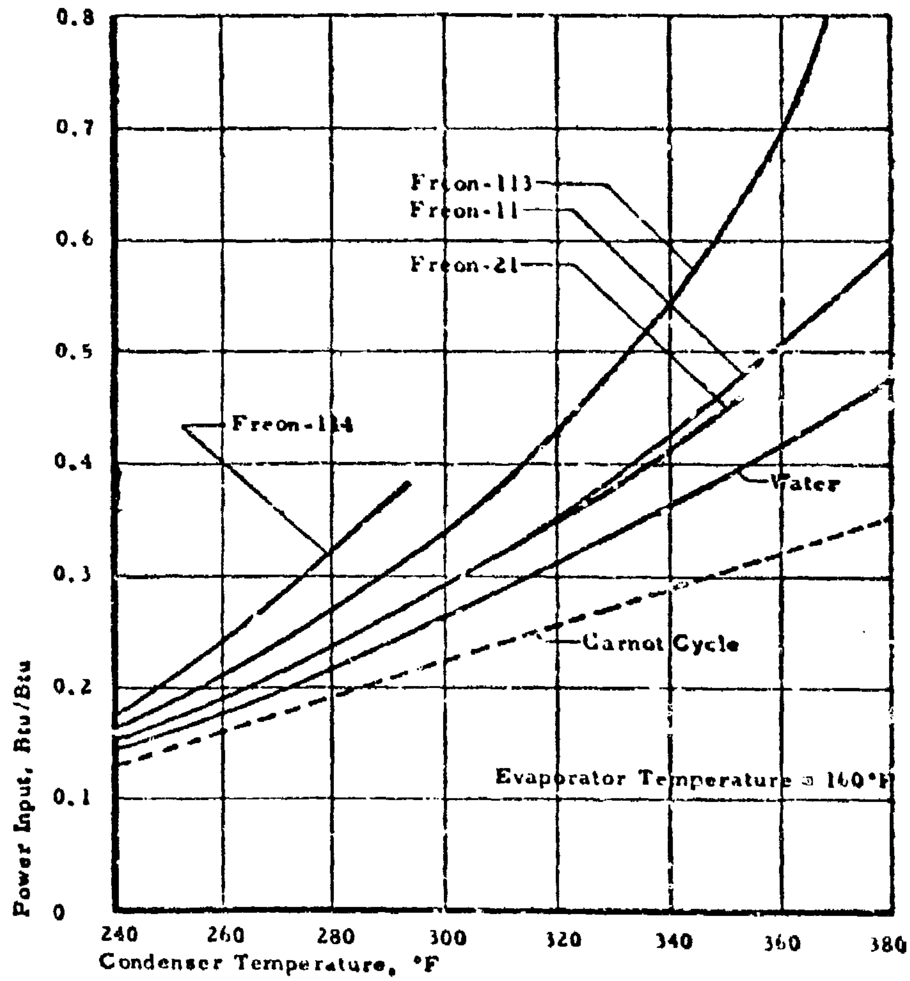


FIGURE 11 POWER INPUT FOR A VAPOR CYCLE ASSUMING IDEAL VAPORS WITH PROPERTIES APPROXIMATING THOSE OF THE SPECIFIED REFRIGERANTS

When checking the suitability of a fluid for a particular application, the following factors should be considered:

1) Critical temperature

2) Freezing point

3) Values of A and B

4) Ideal PI

5) Approximate actual PI (calculated from tables or charts)

6) Pressures and pressure ratios that apply at specified temperatures

7) Cooling effect per pound of refrigerant

8) Volume per unit of refrigeration

The first five factors define the thermodynamic applicability of the fluid; the last three factors serve as an indication of the mechanical design problems, such as compressor type and the size, strength, and weight of components. Some of the more significant properties for a number of refrigerants are listed in table 3 (references 6 and 10 through 14).

7. Use of Water as the Refrigerant

Many of the usual refrigerants cannot be used without an auxiliary air cycle to lower sink temperature for equipment cooling at the higher velocities because of the high total temperatures that prevail. The factors that eliminate such fluids are critical temperature and latent heat of vaporization. Water is a substance that has a high critical temperature (705°F or 1165°R) and a very high latent heat of vaporization (about 1000 Btu/lb at 160°F). The freezing point (32°F) is above some of the minimum temperatures that will probably be encountered, but this would not be a cycle factor during the time when equipment cooling is needed as the temperatures considered are from a minimum temperature of about 150°F. The design of a system which would not be damaged by freezing, e.g., when the aircraft is parked in cold weather, and, further, a system that would automatically thaw out by virtue of heat given off by the equipment to be cooled would not be a particularly difficult engineering problem.

TABLE 3 PROPERTIES OF REFRIGERANTS

	Critical Temp. (°F)	Critical Pressure (psia)	Freezing Point (°F)	Boiling Point (°F)	Latent Heat (Btu/lb) 5°F	Latent Heat (Btu/lb) 48°F	C_{pg} (Btu/lb °F) 88°F	C_{pl} (Btu/lb °F) 88°F
Water	706.1	3226	32	212		[illegible]	0.454	0.998
Dichlorethylene (Dielene)	470.0	795	-70	118				
Methylenechloride (Carrene-1)	421.0	640	-142	103.7	162.1			0.288
Methyl Formate	418.0	607	-148	89.7	[illegible]			
Trichlorotrifluoroethane (F-113)	417.4	495	-31	[illegible]	70.6	65.32	0.154	0.217
Trichloromonofluoromethane (F-11, C-2)	388.4	635	-168	74.7	84	77.16	0.136	0.208
Ethyl Chloride	369.0	764	-218				0.275	
Dichloromonofluoromethane (F-21)	353.3	750	-211	48.0	109.3	98.84	0.169	0.258
Sulphur Dioxide	314.8	1141.5	-99	13.8	169.4		0.154	
Methyl Chloride	289.6	962.2	-144	-10.6	180.7			
Isobutane	272.7	557.1	-229	10.0	159.5			
Ammonia	271.2	1651	-107	-28	565		0.523	
Dichlorodifluoromethane (F-12)	232.7	582	-252	-21.6	69.5	59.35	0.157	0.244
Monochlorodifluoromethane (F-22)	209.8	716	-256	-41.4	93.5	75.63	0.173	0.337
Dichlorotetrafluoroethane (F-114)	294.3					54.26	0.161	0.239
Trichloroethylene			-100	186	103			
Ethylether	377	522	-177.3	94.3	170.7	152		

The critical temperature of water, 705 °F, is well above the maximum total temperature considered in this study (approximately 525 °F) so this is not a major factor. The value of B, $c_{p\ell}/c_{pg}$ is approximately 2 for water, which is large compared with other refrigerants. The value of $A(L_t/c_{pg})$ is approximately 1850 for water at 275 °F, which is very large. Considering these factors together, and in view of the curves of figures 10 and 11, it appears that water would be a very desirable fluid for such high temperature applications. The power requirement for water is much lower at the higher temperatures than for the fluids more commonly used as refrigerants.

The advantages of water as a refrigerant are not surprising when one considers its complete domination in other vapor cycle applications, such as steam heating and steam power generation systems. The thermodynamic properties which make water so attractive in these fields are as appealing for vapor cycle cooling at the higher temperatures. The thermodynamic properties are not even approached by any of the other fluids studied.

The attractive qualities are accompanied by certain factors that create special problems. Among the disadvantages are a rather large variation of vapor pressure with temperature, therefore requiring a relatively high pressure ratio from the evaporator to the condenser. This factor, together with the relatively high c_p/c_v ratio (characteristics of gases with a small number of molecules per atom), results in a gas with a pronounced tendency to superheat when compressed isentropically. If the pressure ratio is high, due to a large temperature difference between the evaporator and condenser, the vapor at the compressor discharge will have a high superheat. In case that temperature is excessive, the compressor must be cooled during compression. A means of controlling the compressor discharge temperature by means of condensate bleed is illustrated schematically in figure 12. The compressor exit temperature is controlled by means of a valve which admits liquid in a fine spray from the condenser to the later stages of compression. The valve is controlled by the compressor discharge temperature. In this way, the superheat will be absorbed in evaporating the liquid. The compressor discharge can be maintained at a predetermined temperature.

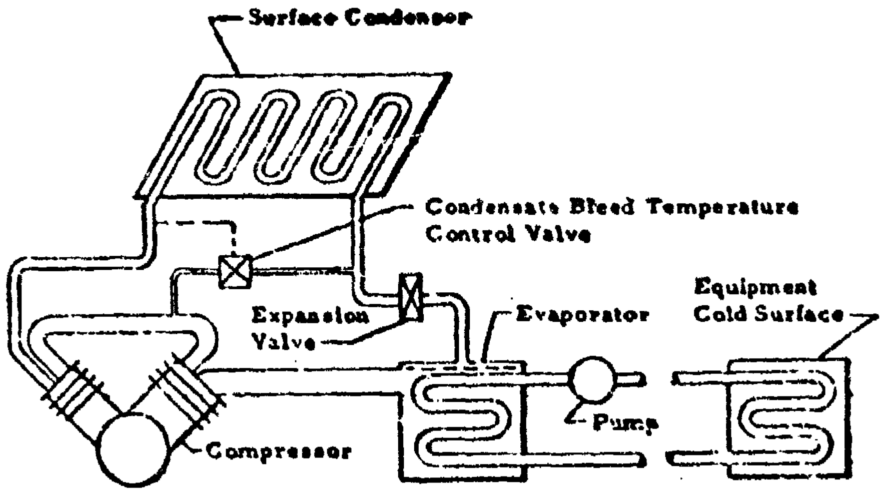


FIGURE 12 WATER VAPOR CYCLE COOLING SYSTEM WITH CONDENSATE BLEED TEMPERATURE CONTROL

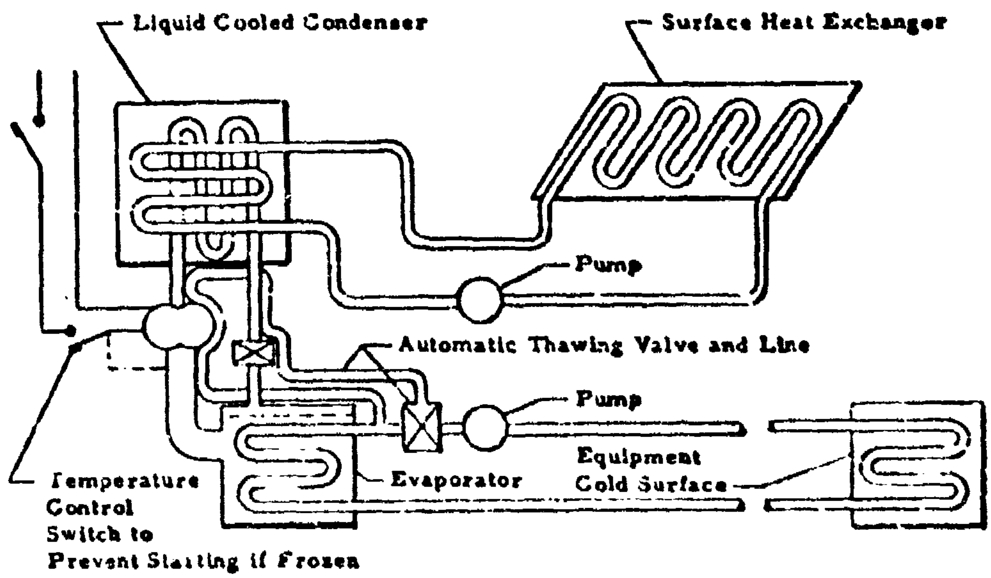


FIGURE 13 WATER VAPOR CYCLE COOLING SYSTEM SHOWING AUTOMATIC THAWING PRINCIPLE

The relatively high freezing point of water is a factor that imposes certain design requirements to eliminate the possibility of damage by freezing and to assure a self-thawing unit so as to be operative when cooling is required. Figure 13 is a schematic diagram pointing out the unit arrangement and design features which could be used to eliminate the adverse effects of freezing. The lines containing water would be made of non-circular sections that would expand without damage if the water freezes. An auxiliary temperature-sensitive switch in series with the regular control switch would prevent starting the unit while frozen but would automatically start the cooling system when the unit becomes ice free. A heat transfer fluid, such as a water-alcohol mixture, that would not freeze would be used to transport the heat from the equipment to the evaporator. The lines carrying the heat transport fluid would be adjacent to, or integral with, any water lines subject to freezing. In this way, the unit would be self-thawing and operative at all times that the equipment approaches the design temperature. The heat of fusion of the water would be utilized for the initial cooling of the heat transfer fluid.

Figures 14 and 15 indicate the power requirement for a vapor cycle cooling system using water and using Freon-11 (one of the better Freons from this point of view). The curves were drawn using actual enthalpy values as taken from tables and charts and, therefore, do not involve the assumption of constant specific heats. Isentropic compression was assumed in each case. This assumption, considering the effect of fluid characteristics as pointed out in a previous part of this report, is probably conservative for water but is nonconservative for Freon-11. The actual difference would, therefore, likely be even greater than indicated by the figure. Freon-11 may be rather unstable at the higher temperatures and would have to be thoroughly investigated before application at the higher temperatures considered.

A study of the figure indicates that water has a slightly lower power requirement than Freon-11 for condenser temperatures up to 275°F. Above 275°F, the difference increases rapidly, the PI for water being about two-thirds that for Freon-11 at a 340°F condenser temperature. Even neglecting instability, the absolute limit for Freon-11 is about 360°F while for water the condenser temperature can go above 600°F. At 360°F, the PI for Freon-11 is approximately the same as at 500°F for water. The deviation from the ideal Carnot cycle is also interesting, e.g., the PI for Freon-11 is approximately twice the Carnot value at 340°F while for water the PI does not double the Carnot value unless the condenser temperature is about 540°F.

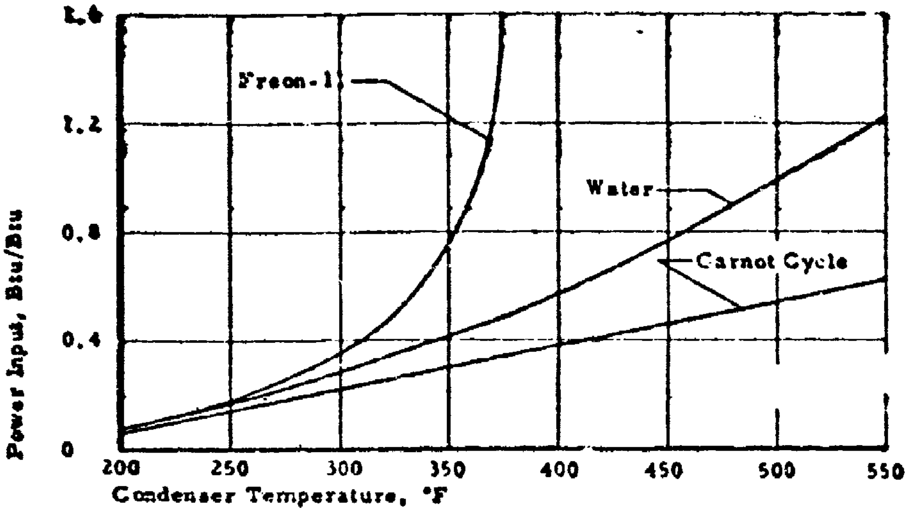


FIGURE 14 POWER INPUT VERSUS CONDENSER TEMPERATURE FOR VAPOR CYCLE COOLING SYSTEM (T_V = 160°F)

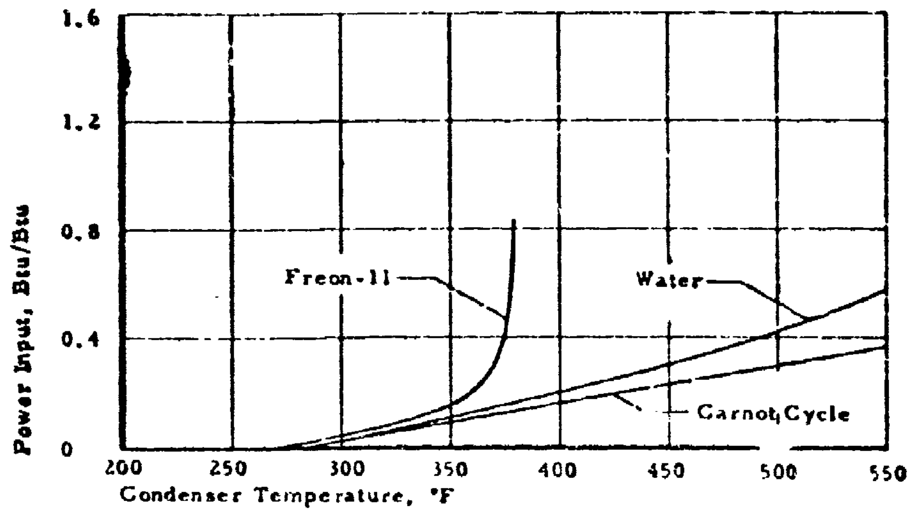


FIGURE 15 POWER INPUT VERSUS CONDENSER TEMPERATURE FOR VAPOR CYCLE COOLING SYSTEMS (T_V = 275°F)

Figure 16 indicates the variation of cooling effect per pound of refrigerant with condenser temperature for water at an evaporator temperature of 160°F and 275°F and for Freon-11 at an evaporator temperature of 160°F. Because of the high latent heat of vaporization of water, the cooling effect is much greater. Figure 17 gives the vapor volume per unit of cooling versus the condenser temperatures for water and for Freon-11. Because of the low density of water vapor at the pressure (about 4.75 psia) that applies for a 160°F evaporator temperature, the volume is high, indicating the need for a relatively large displacement compressor. At a 275°F evaporator temperature, the vapor pressure has increased to about 45 psia and the volume is reduced to approximately one-tenth its value at the 160°F evaporator temperature. Compressor ratios versus condenser temperatures are shown in figure 18 for Freon-11 and in figure 19 for water at evaporator temperatures from 150° to 270°F. The values of cooling effect, volumes of vapor, and pressures that apply for specific conditions determine the type of compressor and the required size, strength, and weight of system components.

C. Mechanical Components of Vapor Cycle Cooling Systems

The mechanical components are the units which contribute most of the weight of the system and because of efficiency and temperature effects have a direct bearing on the power requirements of the cooling system. The components should be designed with due consideration for the refrigerant to be used.

1. Compressors for Vapor Cycle Cooling Systems

The refrigerant in a vapor cycle cooling system must be transferred from the evaporator to the condenser. The vapor leaving the evaporator is at a relatively low pressure and temperature and must be increased to the energy level of the condenser. The additional energy is supplied by means of a compression process. In the conventional refrigeration or heat pump cycle, the difference in energy is supplied by a mechanical compressor. The function of the compressor is to increase the pressure of the refrigerant so as to permit transfer of the fluid from the evaporator to the condenser and to increase the temperature potential so that heat can be transferred to the sink. The compressor is the mechanical unit which compresses the refrigerant and, in effect, pumps the heat to the condenser.

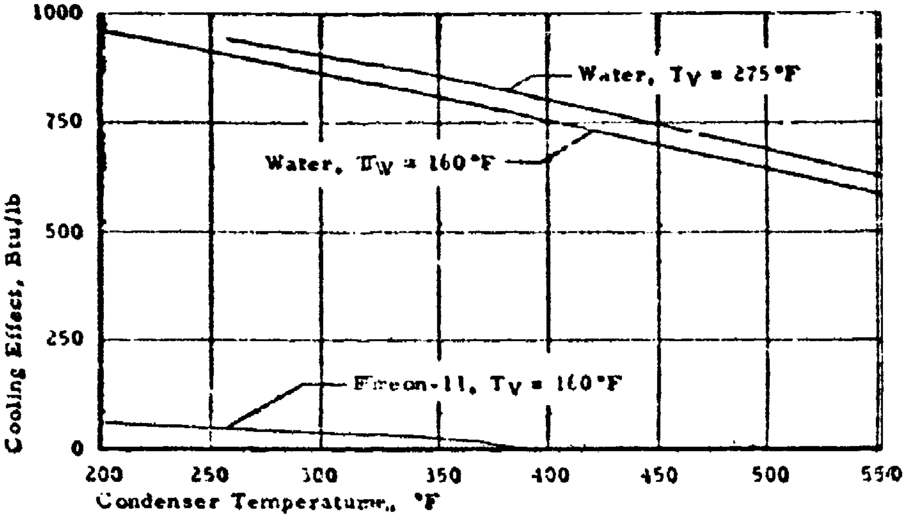


FIGURE 16 COOLING EFFECT PER POUND OF REFRIGERANT VERSUS CONDENSER TEMPERATURE FOR VAPOR CYCLE COOLING SYSTEMS

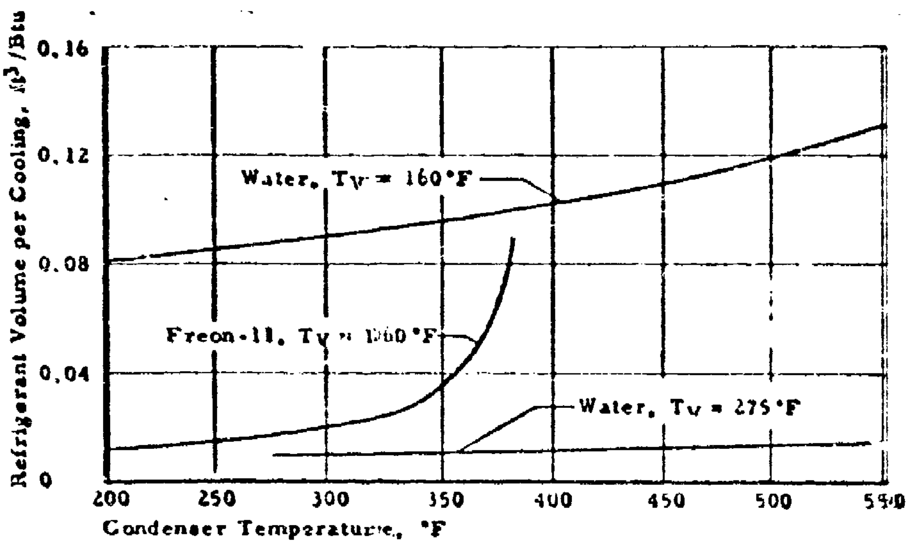


FIGURE 17 REFRIGERANT VOLUME PER UNIT COOLING VERSUS CONDENSER TEMPERATURE FOR VAPOR CYCLE COOLING SYSTEMS

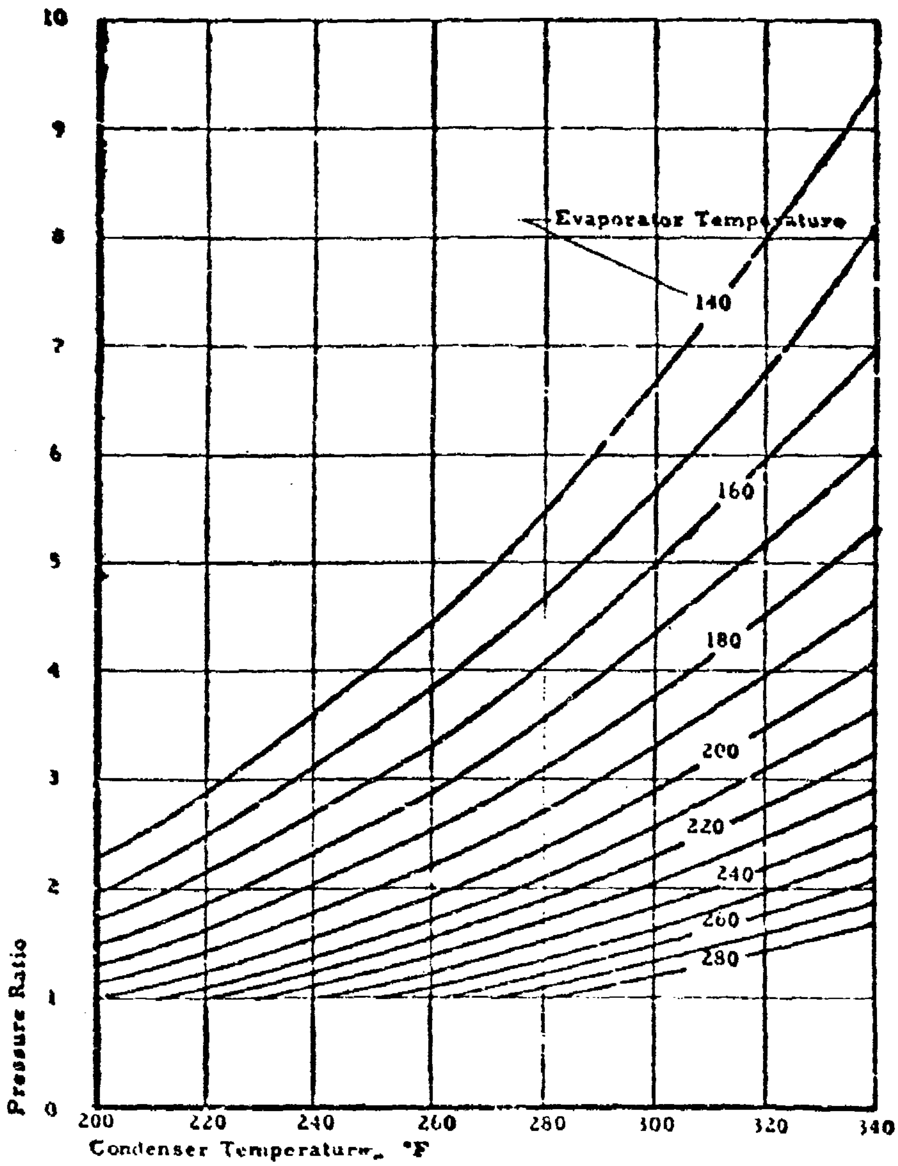


FIGURE 18 PRESSURE RATIO VERSUS CONDENSER TEMPERATURE FOR FREON-11

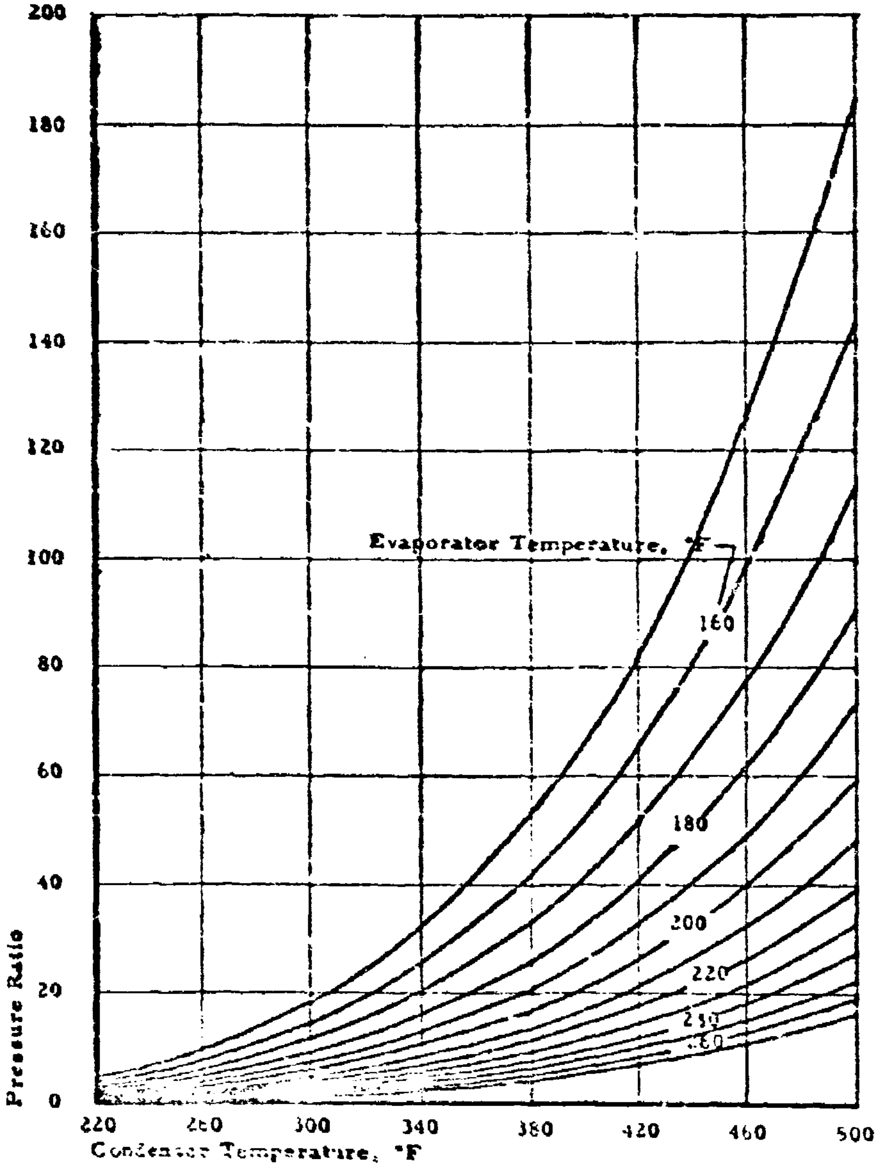


FIGURE 15 PRESSURE RATIO VERSUS CONDENSER TEMPERATURE FOR WATER

The basic considerations that determine the type of compressor that can be used for a vapor cycle cooling system are:

1) The refrigerant properties

2) The required condenser temperature

3) The required evaporator temperature

The refrigerant vapor pressure at the condenser and at the evaporator temperature defines the required pressure ratio and the working pressure. The specific volume, latent heat, and specific heats at the applicable temperatures define the volume and weight of fluid that must be pumped and the required energy input. The effects of such fluid properties are discussed in the preceding part of this report. The molecular weight or density of the vapor, together with the volume flow and pressure ratio required, are the major factors which dictate the type of compressor which can be used for a particular application. There are two general types of compressors that have applications for vapor compression, dynamic and positive displacement types.

The refrigerant in a vapor cycle system can be compressed by dynamic type compressors in which the compression is obtained by virtue of a change in velocity of the vapor. Examples of this type are axial flow and centrifugal compressors. The former would not likely have any application in vapor cycle cooling systems. Centrifugal compressors are widely used for vapor compression. The compression ratio that can be attained with centrifugal units depends on the tip speed, i.e., the rotational velocity, and the radius; and on the density of the vapor. Centrifugal compressors are therefore usually used to compress relatively dense vapors and in applications where the pressure ratio (per stage) is not too great. Centrifugal units are particularly attractive for relatively large volume flow rates. This type of compressor is discussed and analyzed in references 1 and 9.

Positive displacement compressors are widely used both in conventional piston-type units and in many variations of rotary type units. Reciprocating units require very careful design to reduce the weight of this type of unit. The usual somewhat large weight is primarily a result of the relatively low rotational speeds usually used with this type of compressor. Rotational speeds are limited by the reciprocating parts and by limitations imposed by the conventional reed-type valves. By utilizing light weight parts and possibly positive action valves, poppet or rotary type, reciprocating piston compressors could possibly be designed that would be competitive on a weight basis with other types of compressors. In general, reciprocating compressors are somewhat more efficient than other positive displacement units.

The positive displacement rotary type of compressor appears very promising for lightweight aircraft cooling systems. The mechanical simplicity and high rotational speeds that can be used with units of the Lysholm type result in very light weight and high capacity units. The weight advantage of such units is partially offset by the lower efficiency, typically about 65%, as compared with efficiencies of approximately 80% for reciprocating compressors.

For the purposes of an analytical study, it is desirable to define a relationship between the weight of a compressor and the operational factors such as refrigerant flow, pressure ratio, volumetric efficiency, number of stages of compression, etc. A study of a number of compressors made by various manufacturers indicates that the weight of a compressor can be approximated by an empirical formula of the type

$$W_C = \left(A + \frac{B\ V_R}{\eta_v}\right)(n)^a \qquad (52)$$

For a reciprocating compressor designed for aircraft application, the constants were evaluated to secure an empirical equation which will approximate actual units for the displacement range from 2 to 40 cfm.

$$W_C = \left(4 + \frac{0.5 V_R}{\eta_v}\right) n^{1/2} \qquad (53)$$

The weight of positive displacement rotary compressors of advanced design can be approximately defined by a similar equation for the given displacement range,

$$W_C = \left(3 + \frac{0.1 V_R}{\eta_v}\right) n^{1/2} \qquad (54)$$

The constants in the above equations can, of course, be selected so as to approximate other types of compressors. For a particular application, the actual weight should be used if it is known.

The power supply system for a vapor cycle system must furnish power for vapor compression and to pump the transfer fluid through the lines from the equipment to the evaporator and in some cases from the condenser to a heat exchanger. The fundamental effects of the power supply system are a weight addition and a power requirement.

The required refrigerant volume flow rate (V_R) at evaporator conditions per Btu and per kw is plotted versus evaporator temperature for Freon-11 and for water in figures 20 and 21, respectively. The required compressor displacement is equal to the refrigerant volume flow divided by the volumentric efficiency (η_v).

The power input for the compressor is equal to the cooling load times power input divided by the compressor efficiency. With the cooling load expressed in kilowatts (kw), the required power in horsepower (HP) is

$$HP = \frac{kw\ PI}{0.746\ \eta_c} \tag{55}$$

The power for circulating the heat transport fluid is dependent on the flow rates, line length, diameter, and on the fluid properties. The heat transport part of a cooling system is discussed in section IV and in appendix II of this report. The power can be supplied by an electric system, a pneumatic system, or a hydraulic system.

The weight of an electric power supply system includes the weight of the motor, the additional weight necessary for increased generator capacity, and the weight of wiring and controls.

The weights of various sizes of typical electric motors made by different manufacturers have been investigated. The weight of typical high speed aircraft type a-c motors are plotted in figure 22. A good approximation to the weight of high speed a-c motors for the 1 to 10 HP range is given by the linear equation

$$W = 5 + 1.5\ HP \tag{56}$$

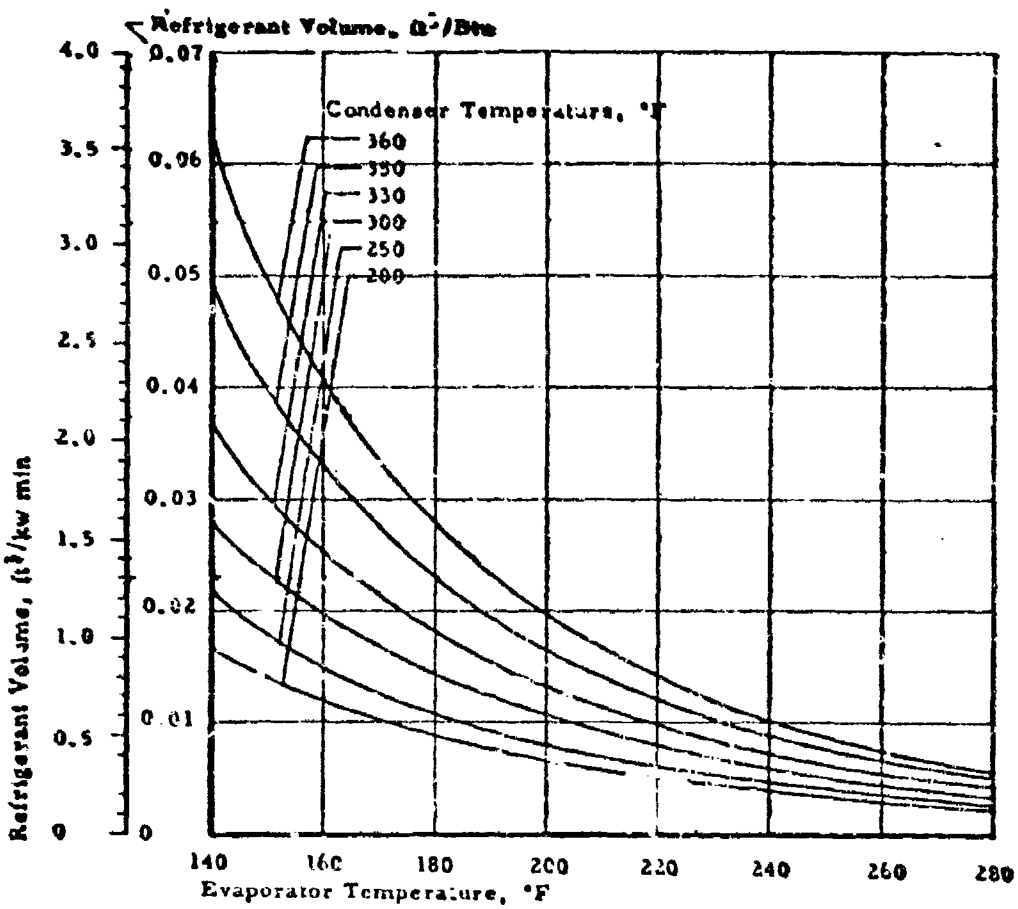


FIGURE 20 REFRIGERANT VOLUME VERSUS EVAPORATOR TEMPERATURE FOR FREON-11 VAPOR CYCLE COOLING SYSTEMS

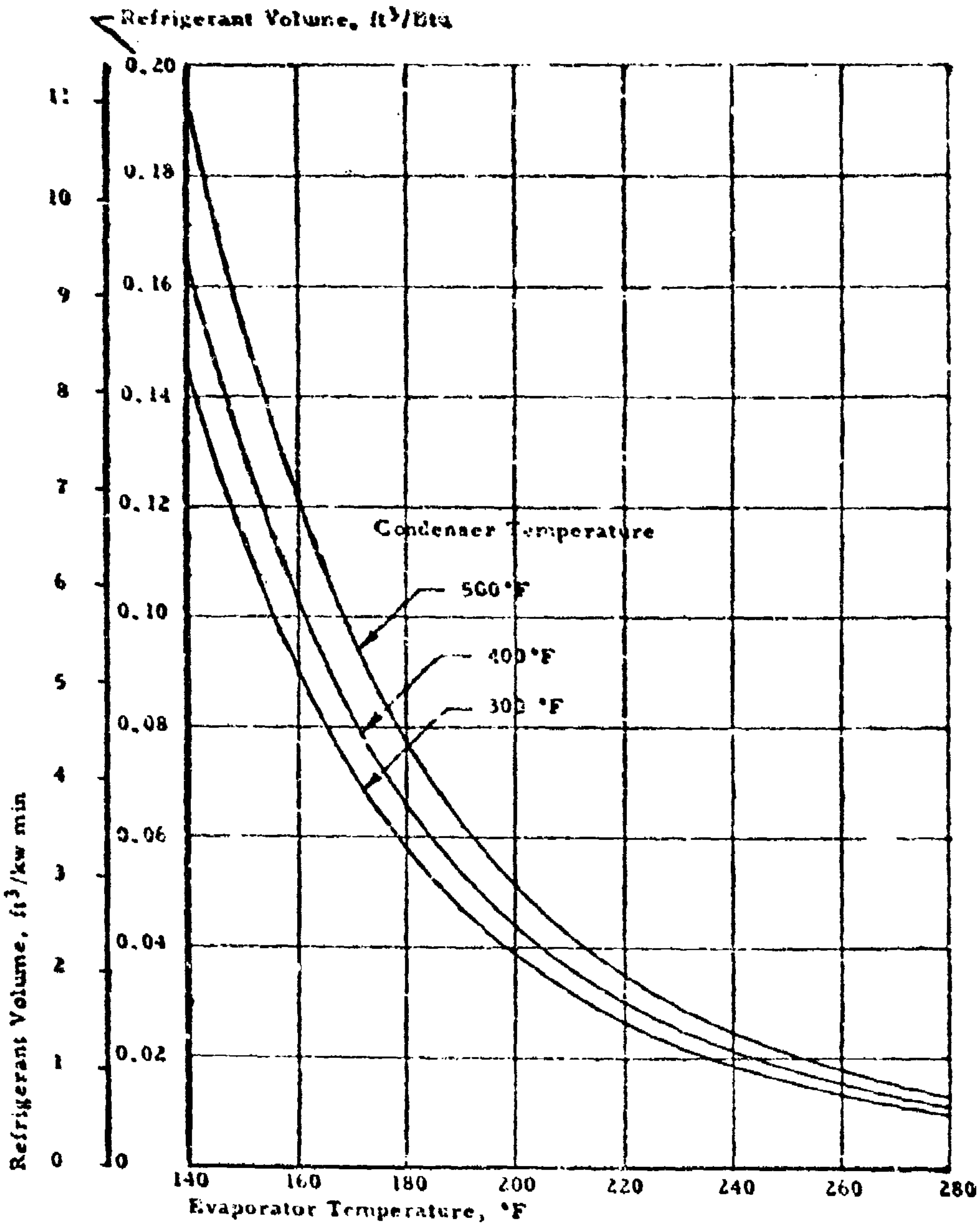


FIGURE 21 REFRIGERANT VOLUME VERSUS EVAPORATOR TEMPERATURE FOR WATER VAPOR CYCLE COOLING SYSTEMS

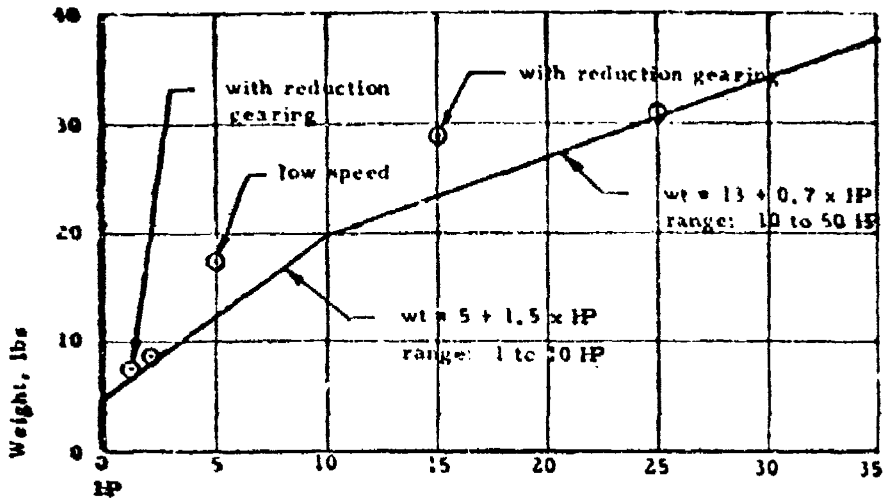


FIGURE 22 WEIGHT OF HIGH SPEED A-C MOTORS
(200 VOLTS, 400 CPS, 3-PHASE)

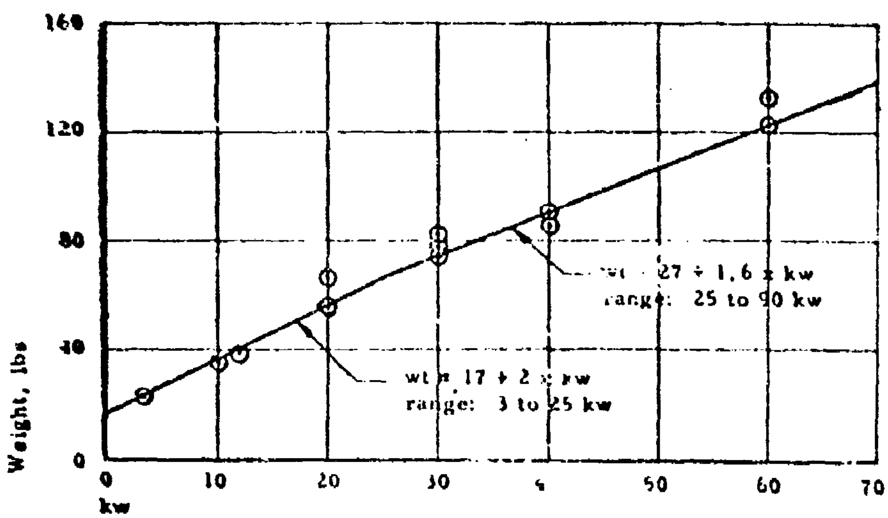


FIGURE 23 WEIGHT OF A-C GENERATORS
(200 VOLTS, 400 CPS, 3-PHASE)

The weight of larger, high speed motors, from 10 to 50 HP can be approximated by the equation

$$W = 13 + 0.7\ HP \tag{57}$$

The weights of typical a-c electric generators are plotted in figure 23. The linear approximation to the weight of generators in the range from 3 to 25 kw is given by the equation

$$W = 17 + 2\ kw$$

The weight of generators for the range 25 to 90 kw can be approximated by the equation

$$W = 27 + 1.6\ kw \tag{58}$$

In determining the generator weight to be charged to the cooling system, it is assumed that the aircraft will have a generator and that the cooling system will require an increase in the generator capacity and, therefore, will be charged with the additional weight. It is assumed that the actual generator is in the larger range and, to allow for motor efficiency and line losses, it will be assumed that each horsepower requires one kilowatt generator capacity. The generator weight for RC units charged to the cooling system is then approximately defined in terms of required horsepower by the equation

$$W = 1.6\ HP \tag{59}$$

The weights of typical d-c motors and generators are plotted in figures 24 and 25, respectively. A comparison of the a-c and d-c data indicates a relatively large weight advantage for the a-c units.

The power input versus condenser temperature for the entire range of evaporator temperatures are plotted in figure 26 for vapor cooling cycles using Freon-11 and in figure 27 using water. It should be noted that on the curves for the water cycle, the scale of the horizontal axis is 40° between each vertical line as compared with 20° for the Freon cycle. Thus the power input for the water cycle increases much more slowly with increasing condenser temperature than it does for the Freon-11 cycle. The values as plotted are dimensionless (e.g., Btu/Btu or kw/kw) and are plotted assuming isentropic compression neglecting valve or tubing pressure drops. The effect of inlet and outlet pressure drops are illustrated by the curves of figures 28 and 29, respectively.

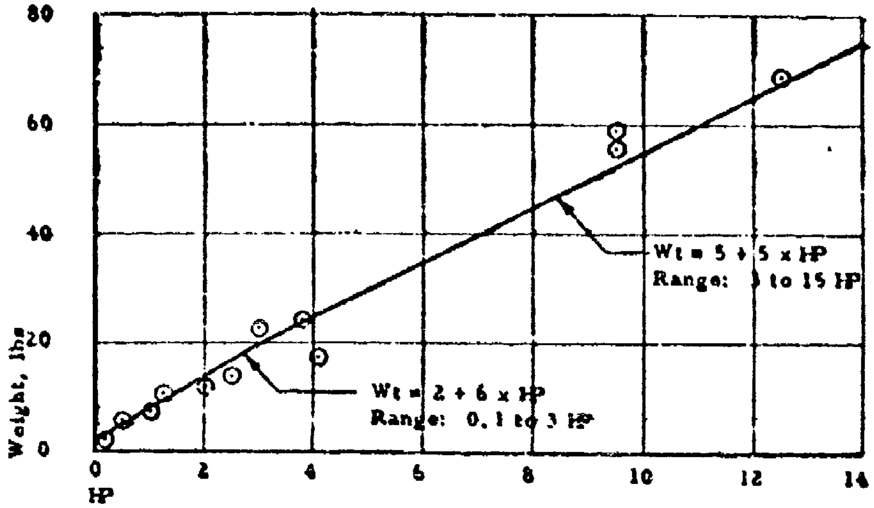


FIGURE 24 WEIGHT OF D-C MOTORS, 27 VOLTS

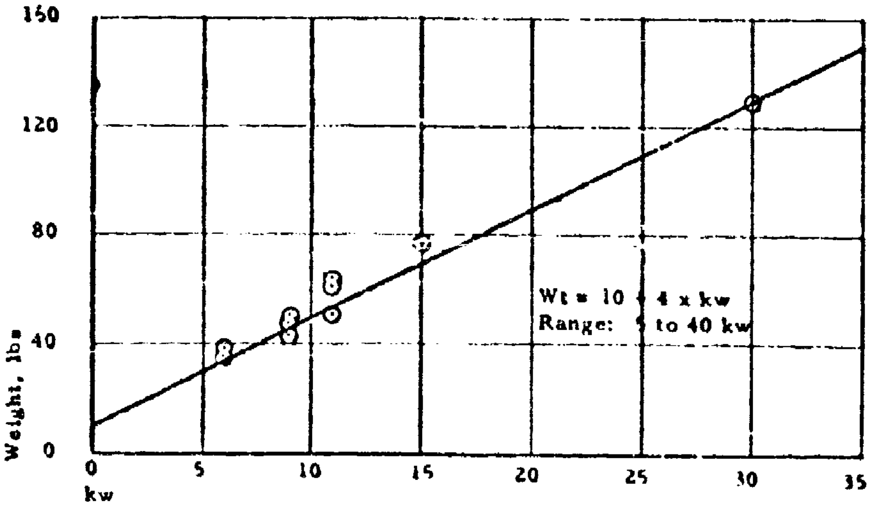


FIGURE 25 WEIGHT OF D-C GENERATORS, 30 VOLTS

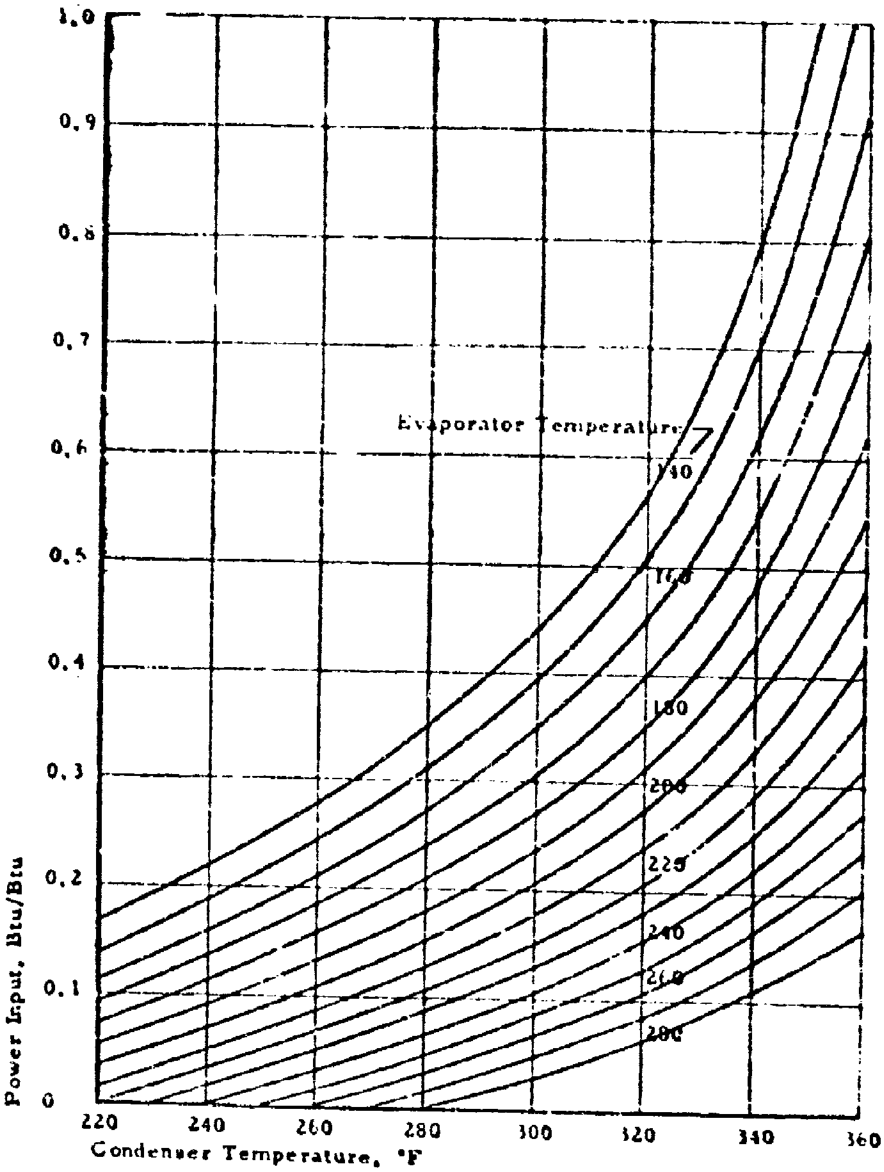


FIGURE 26 POWER INPUT VERSUS CONDENSER TEMPERATURE FOR A FREON-11 VAPOR COOLING CYCLE

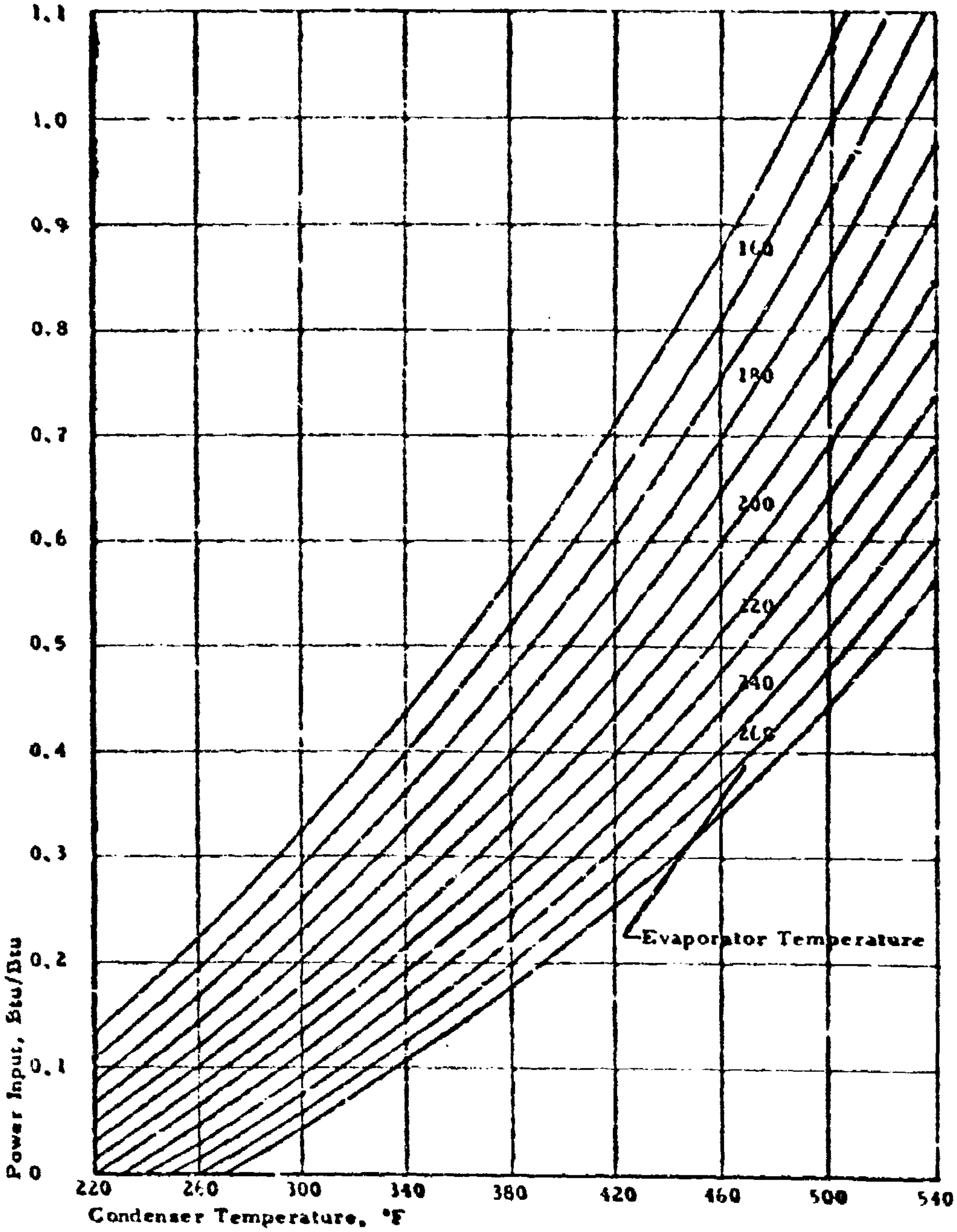


FIGURE 27 POWER INPUT VERSUS CONDENSER TEMPERATURE FOR A WATER VAPOR COOLING CYCLE

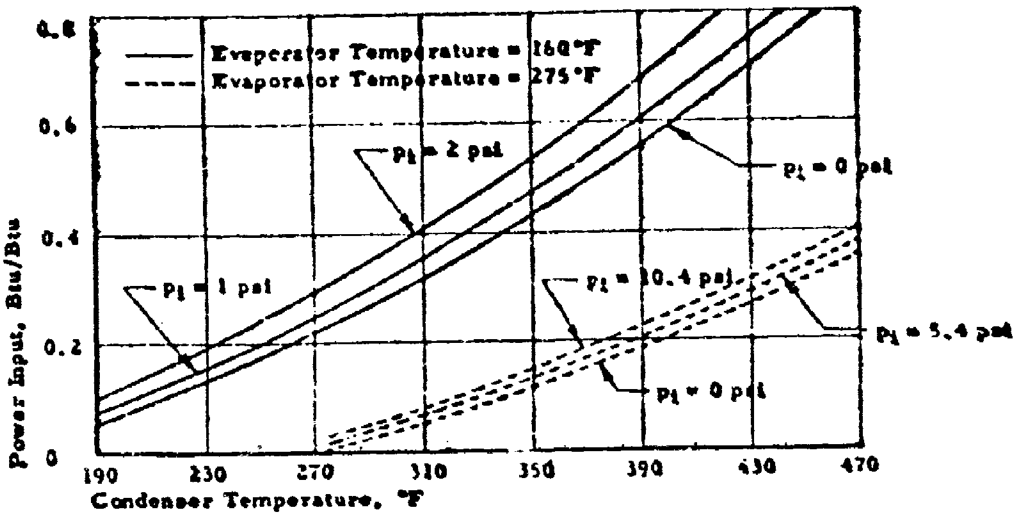


FIGURE 28 EFFECT OF INLET PRESSURE DROP FOR WATER VAPOR CYCLE COOLING SYSTEMS

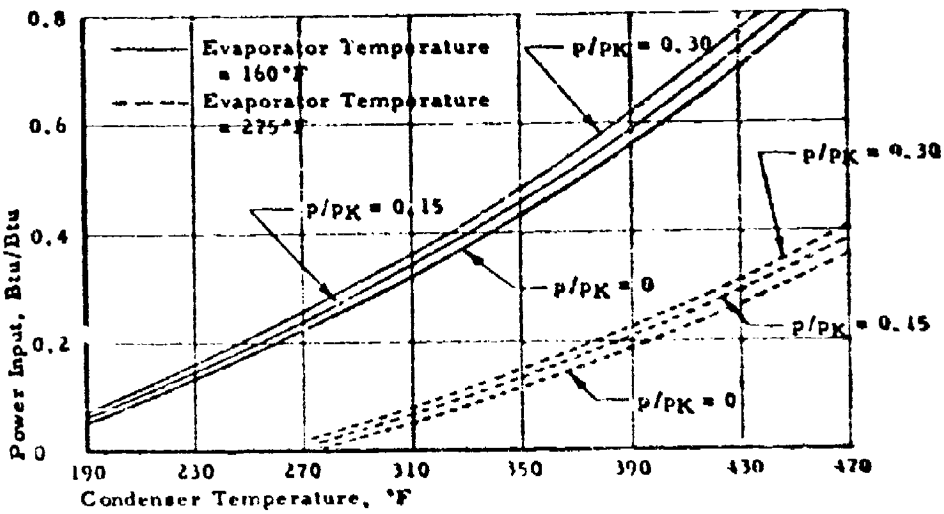


FIGURE 29 EFFECT OF COMPRESSOR OUTLET PRESSURE DROP FOR WATER VAPOR CYCLE COOLING SYSTEMS

At the lower temperatures and the smaller differences between condenser and evaporator temperature, the power input, assuming isentropic compression, is within about 10% for Freon-11 and for water. For all conditions in the range considered, water exhibits a lower PI than does Freon-11, about 70% of the PI at a 150°F evaporator and 300°F condenser temperature. Water requires less than half the power at a 150°F evaporator temperature and 360°F condenser temperature. Freon-11 has a critical temperature, 388°F, so would not be applicable for condenser temperatures greater than about 360°F, or perhaps much lower from stability considerations. Water with a critical temperature of 705°F and being very stable could be used with condenser temperatures of 500° to 600°F or greater, but of course the condensing pressures would then be high (680 psia at 500°F, 1540 psia at 600°F). All the curves were drawn assuming isentropic compression and neglecting valve pressure drops. In the case of water vapor which has a marked tendency to superheat with isentropic compression, the power input would decrease somewhat if the compressor were cooled, thus reducing the outlet temperature. Such cooling of the compressor is essential to prevent excessive temperatures when high pressure ratios are involved.

In a vapor cycle cooling system, the major fundamental effects are weight addition and power consumption and, in some cases, a drag effect. The compressor and power source are the units which directly contribute most of the weight of the system and are the major power-consuming components. The design of the evaporator and condenser have a direct effect on the volume and on the compression ratio required and thus affect the compressor design. Because of the significance of the compressor in the complete system, design which reduces its weight and power is a very fruitful field for system weight and power reduction. In particular, an increase in the evaporator temperature will reduce the power input and decrease the displacement and weight of the compressor because a higher vapor pressure will reduce the specific volume of the fluid at the compressor inlet and also increase the pressure and, therefore, reduce the required pressure ratio from the evaporator to the condenser. A reduction in the condenser temperature will result in lower compressor discharge pressures and therefore also reduce the power input.

2. Evaporators

When aircraft equipment is cooled by a vapor cycle cooling system, the heat dissipated by the equipment is absorbed in evaporating the refrigerant. In most cases, the heat from the equipment would likely be transported from the equipment to the evaporator by means of a heat transport fluid. The transport of the neat is discussed in section IV of this report. In this case, the evaporator is a liquid-to-boiling fluid heat exchanger. Ideally, to secure a minimum of temperature drops, the evaporator should be located at or in the equipment, thus cooling the equipment directly. However, to secure a compact unit, to eliminate long lines for conducting the vapor from the equipment to the compressor, and considering the relatively high heat transfer coefficients that apply for liquids flowing in tubes, the use of a heat transport fluid is usually justified.

Heat transfer coefficients that apply for evaporative heat transfer for the refrigerants considered in this study are assumed constant and assigned typical values as given in the literature. Reference 5 gives the value of evaporative heat transfer coefficients for water as 500 to 1000 Btu/hr ft^2 °F. Reference 1 gives values in the range from 300 to 600 Btu/hr ft^2 °F as typical for evaporating Freon-11.

The following factors must be considered as they affect evaporator design and total weight and power requirements of a cooling system:

1) Surface area for evaporative heat transfer and for transfer fluid heat transfer - In general, a large area will result in heavier evaporator, higher evaporating temperature and therefore lower compressor displacement, weight, and power requirements.

2) Transfer fluid velocity and mass flow rate - In general, high velocity and mass flow rate will result in a high heat transfer coefficient, small temperature rise through evaporator, and permit higher evaporating temperatures, reducing compressor size and power but requiring greater transfer fluid pumping power.

3) Evaporator design should be aimed toward high heat transfer coefficients for both the boiling refrigerant and the flowing heat transport fluid. The weight and volume should be minimized for the required heat transfer area and strength.

The following evaporator discussion is based on the assumption that the cooling load is to be extracted from a liquid which has absorbed the heat dissipated by the equipment. The evaporator is then a liquid to tube wall, to boiling liquid heat exchanger.

The high heat transfer coefficients that apply in the evaporator indicate that the temperature difference in this particular type heat exchanger can be relatively small, and also that the heat transfer areas and weight of the unit can be kept quite low. The required area will vary inversely with the temperature difference. The optimum values are governed by weight considerations, by the required evaporating temperature, and by the power requirements. The weight associated with the evaporator is the weight of the tubes, or heat transfer surfaces, the shell, the heat transfer fluid, and the refrigerant. For a particular configuration, such weight factors can be defined in terms of the specific weights and the required heat transfer area.

The power required to circulate the heat transport fluid will depend on the flow rate, on the tube geometry, and on the fluid viscosity. In addition to the power required to circulate the fluid, the evaporator temperature will influence the vapor compression power requirement because of the variation of the vapor specific volume and required compression ratio with the evaporating temperature.

The evaporating temperature can be determined by assuming that the heat, or cooling load, has been absorbed by a transfer fluid and must be removed from the liquid at an evaporator inlet temperature which is equal to the equipment exit temperature (T_{E_e}). This temperature is dependent on the operating limitations of the equipment being cooled.

For this analysis, the heat transfer fluid will be assumed to be flowing inside a tube with the evaporating refrigerant on the outside. Other details of the configuration need not be defined at this point. The convective inside and outside heat transfer coefficients, h_i and h_o, will be assumed constant (or effective average values used).

Assuming the inside and outside heat transfer areas are equal, and neglecting the small temperature drop through the tube wall, the heat transfer in a length dx of an evaporator tube is

$$dQ_V = \pi d \left(\frac{h_o h_i}{h_o + h_i} \right) (T_f - T_V)\, dx$$

$$= U \pi d (T_f - T_V)\, dx \tag{60}$$

The temperature of the fluid (T_ℓ) will vary along the tube as the heat is dissipated. The heat loss can also be expressed in terms of fluid temperature change

$$\frac{dQ_V}{dx} = \left(\frac{\pi d^2}{4}\right) \cdot 3600\, V_g \rho c_p \frac{dT_\ell}{dx}$$

$$= 3600\, wc_p \frac{dT_\ell}{dx} \qquad (61)$$

Then, from equations (60) and (61),

$$\frac{dT_\ell}{T_\ell - T_V} = \frac{U\pi d}{3600\, wc_p}\, dx \qquad (62)$$

Integrating between inlet and outlet temperature ($T_{\ell i}$ and $T_{\ell o}$) and along length (ℓ),

$$\ln\left(\frac{T_{\ell o} - T_V}{T_{\ell i} - T_V}\right) = \frac{-UA}{3600\, wc_p} \qquad (63)$$

Integrating equation (61) gives

$$Q_V = 3600\, wc_p\, (T_{\ell i} - T_{\ell o}) \qquad (64)$$

The term $3600\, wc_p$ can be eliminated from equation (64) by substituting a value determined from equation (63).

$$Q_V = \frac{UA\,(T_{\ell i} - T_{\ell o})}{\ln\left[(T_{\ell i} - T_V)/(T_{\ell o} - T_V)\right]} \qquad (65)$$

Equation (65) gives the evaporator heat transfer in a form similar to the usual convective heat transfer equation in terms of a logarithmic mean temperature difference.

The temperature at which the refrigerant evaporates (T_V) is very significant as this temperature is one of the factors that determines the cycle temperature difference, thus affecting the power input and refrigerant flow requirement. The refrigerant evaporating temperature also determines the specific volume of the vapor at the compressor inlet. These factors determine the necessary compressor displacement. The actual refrigerant evaporating temperature must be optimized considering weights of compressor, evaporator, and distribution system and also considering the refrigerant compressor and the heat transfer fluid pump power requirements.

The required evaporating temperature (T_V) can be determined as follows:

From equation (65)

$$\ln\left(\frac{T_{fi} - T_V}{T_{fo} - T_V}\right) = \frac{UA}{Q_V}(T_{fi} - T_{fo}) \tag{66}$$

Substituting the value of Q_V from equation (64), taking the anti log, and solving for T_V,

$$T_V = \frac{T_{fi} - T_{fo}\, e^{\frac{UA}{3600\, wc_p}}}{1 - e^{\frac{UA}{3600\, wc_p}}} \tag{67}$$

The transfer fluid outlet temperature T_{fo} can be expressed in terms of the inlet temperature, the evaporator load, the tube size, and the flow rate.

$$T_V = T_{fi} - \frac{Q_V}{3600\, wc_p}\left[\frac{e^x}{(1 - e^x)}\right] \tag{68}$$

where $x = \frac{UA}{3600\, wc_p}$

The evaporating temperature must be lower than the transfer fluid temperature but should be as high as is practical. As T_V increases, the vapor pressure increases; therefore, the power required to compress the vapor will decrease as will the specific volume of the vapor and the required compressor displacement. The lower compression power and weight will of course require greater evaporator weight and greater transfer fluid pumping power. The design value of T_V should be selected considering all of the effects.

Calculations indicate that the weight of an evaporator of the type illustrated in figure 31 can be approximated by an empirical equation of the form

$$W_V = \frac{C \times Q_V}{T_{Ee} - T_V} \tag{69}$$

If the cooling load is expressed in kilowatts, the constant C will have a value of about 10 for an efficiently designed evaporator. The evaporator weight is then

$$W_V = \frac{10\ kw}{T_{Ec} - T_V} \tag{70}$$

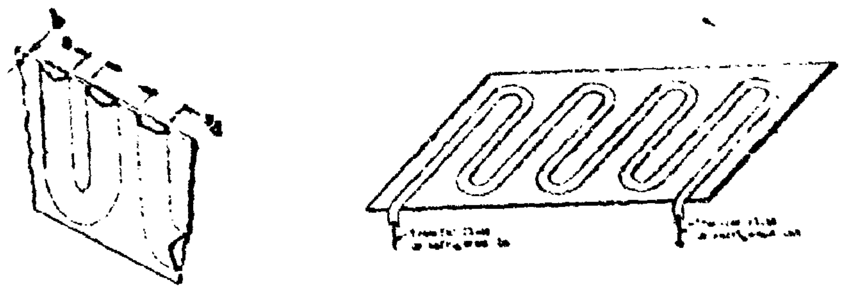
In evaluation calculations, an optimum value of $T_{Ec} - TV$ was determined considering the variation of compressor and power supply system weight so as to secure a minimum total weight (including a power equivalent weight).

3. Condenser

The condenser of a vapor cycle cooling system must dissipate the cooling load plus the heat generated during compression of the vapor. That heat must be given off to a heat sink, most likely the atmosphere. The amount of heat that can be transferred through a given area is dependent on the area, the temperature difference, and the heat transfer coefficient. The area should be as small as possible to minimize weight. The temperature difference should be as small as possible so as to reduce the work of compression.

At the higher altitudes considered in this study, the density of the air is very low resulting in very low heat transfer coefficients unless the velocity of airflow is very high.

A surface-type condenser is shown in figure 30, a ram air cooled condenser in figure 32, and a liquid cooled condenser in figure 33. The surface-type condenser provides a means of securing a reasonably high heat transfer coefficient even at the high altitudes because of the extremely high velocities that apply for heat transfer. An additional advantage of this type of heat exchanger is that the effective sink temperature is the recovery temperature and not the total temperature of the air. This is a very significant factor at the high velocities. This type is also particularly attractive in that, being a part of the surface of the aircraft, there would be a relatively small increase in drag to be charged to the air side of this type of condenser.



a. Typical Section of Unit

b. Schematic

FIGURE 30 SURFACE HEAT EXCHANGER OR CONDENSER

a. Typical Unit

b. Schematic

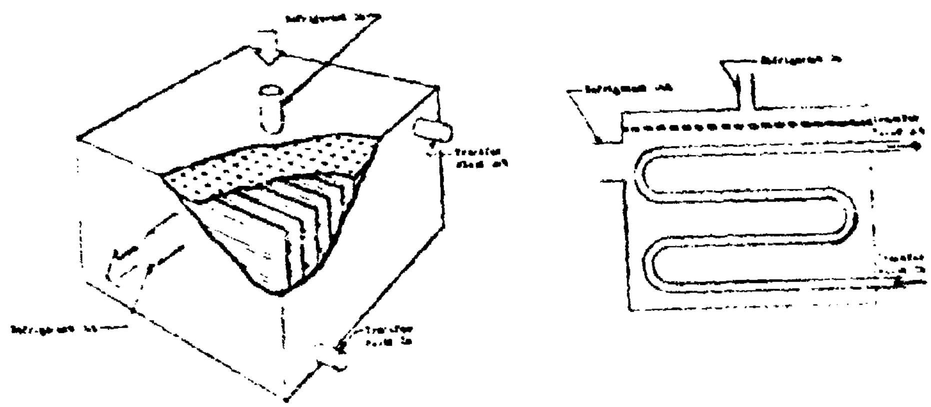


FIGURE 31 EVAPORATOR FOR LIQUID HEAT TRANSPORT FLUID

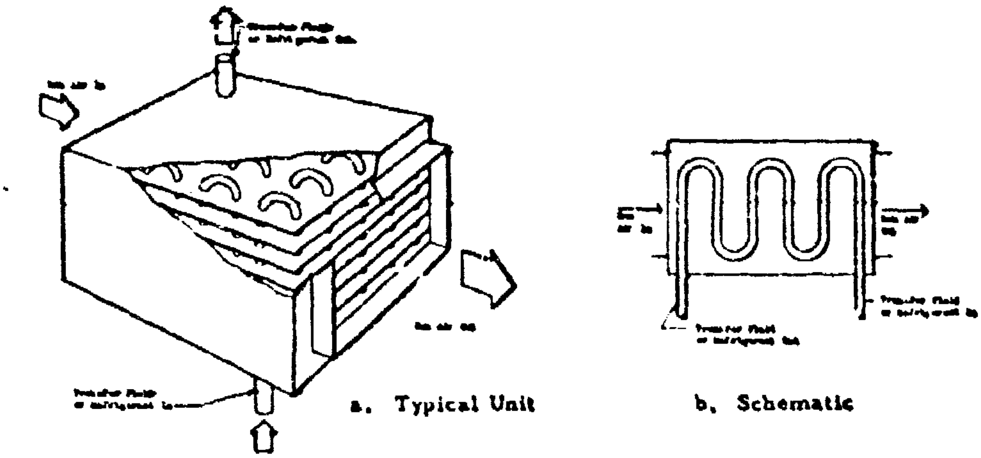


FIGURE 32 RAM AIR COOLED CONDENSER OR HEAT EXCHANGER

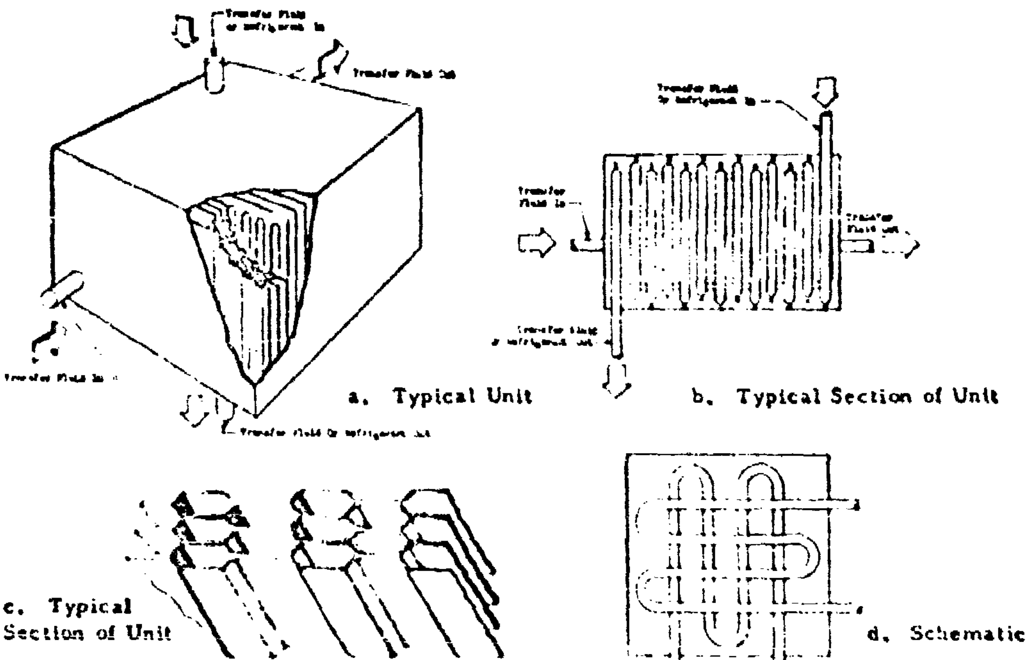


FIGURE 33 LIQUID COOLED CONDENSER OR HEAT EXCHANGER

The major effect of a surface condenser or heat exchanger is the additional weight imposed on the aircraft. The surface heat exchanger will also impose an increase in drag. The drag increase is caused by the temperature change along the surface of the aircraft. The change in temperature results in an increase in the local heat transfer coefficient and also in the skin friction. A flat plate with an unheated area followed by an area that is heated will have a larger local heat transfer coefficient than a plate heated over the entire area (reference 15). The local heat transfer coefficient (h') for the plate with the unheated area can be expressed in terms of the local heat transfer coefficient (h) that would apply if the plate were uniformly heated (for laminar flow)

$$h' = h \frac{1}{\left[1 - \left(\frac{x_o}{x}\right)^{3/4}\right]^{1/3}} \qquad (71)$$

In equation (71), x_o is the distance from the leading edge to the heated area and x is the distance from the leading edge to a point in the heated area. Because of heat conduction considerations, the temperature increase is not a step change, and the infinite increase in heat transfer coefficient indicated by equation (71) when $x = x_o$ is of course not valid. The determination of the drag caused by a surface heat exchanger would require an expression indicating the change in skin friction and also rather detailed assumptions and calculation of the skin friction and drag for the case without a temperature variation. Such a drag analysis has not been made in the present study. Neglecting the increase in the heat transfer coefficient is a conservative assumption which will tend to offset (from an evaluation standpoint) the increase in drag which has not been analyzed in this study. The heat transfer surface should be located in a region of turbulent flow so that the temperature change will not cause transition at a point that would normally be in laminar flow.

Assuming the surface is in a region of fully developed turbulent flow, neglecting the increase in heat transfer coefficient and in skin friction will not introduce a serious error.

At the lower flight speeds, the heat transfer coefficient decreases; however, the recovery temperature will also decrease so the temperature difference increases. The greater temperature difference will more than offset the heat transfer coefficient decrease. The dissipation of a given amount of heat will therefore be most critical at the maximum velocity that applies for the specified flight limits.

With the above assumptions, the external heat transfer coefficients that apply for various altitudes are plotted versus velocity in figure 4. Curves of constant recovery temperature are plotted on the altitude-Mach number axis in figure 5.

The heat transfer coefficients associated with condensation of vapors are usually very high. Consequently, the temperature drop and the area required to transfer the heat from the vapor to the atmosphere will be primarily determined by the air side heat transfer coefficients. The area that is required to transfer a given amount of heat can be determined assuming that the part of the surface with a direct conductive path from the tube is at a constant temperature and neglecting any heat transfer to the inside, which would likely be an insulated surface. The effect of tube spacing and other characteristics can be determined from such considerations.

The heat conducted along the surface between the tubes (figure 30) is

$$Q = -\,k\,b\,\ell\,(dT/dx) \tag{72}$$

The change in Q in a distance dx is equal to the convective heat transfer

$$dQ = h\,\ell\,(T - T_r)\,dx \tag{73}$$

or

$$(dQ/dx) = h\,\ell\,(T - T_r) \tag{74}$$

Differentiating equation (72) with respect to x,

$$(dQ/dx) = -\,k\,b\,\ell\,(d^2T/dx^2) \tag{75}$$

Subtracting equation (74) from equation (75)

$$(d^2T/dx^2) + (h/k\,b)\,(T - T_r) = 0 \tag{76}$$

The solution of this differential equation is

$$T = T_r + C_1\,e^{\sqrt{\frac{h}{kb}}\,x} + C_2\,e^{-\sqrt{\frac{h}{kb}}\,x} \tag{77}$$

where C_1 and C_2 are constants of integration.

Determining the constant from the conditions that at $x = 0$, $T = T_W$ and at $x = s/2$, $dT/dx = 0$ and simplifying

$$T = T_r + \frac{\cosh\left[\sqrt{h/kb}\,(s/2 - x)\right]}{\cosh\left[\sqrt{h/kb}\; s/2\right]}\,(T_W - T_r) \qquad (78)$$

Equation (78) defines the temperature at a distance x from the tubes. The heat transferred through the area between the tubes is

$$Q = \frac{2h\ell\,(T_W - T_r)}{\cosh\left[\sqrt{h/kb}\; s/2\right]}\int_0^{s/2} \cosh\left[\sqrt{h/kb}\,(s/2-x)\right] dx \qquad (79)$$

Integrating equation (79)

$$Q = 2h\ell\,(T_W - T_r)\,\sqrt{kb/h}\;\tanh\left(\sqrt{h/kb}\; s/2\right) \qquad (80)$$

A check of equation (80) indicates that the tube spacing, s, should be selected depending on the surface heat transfer coefficient, the conductivity of the material, and the thickness. For example, if s is such that $\sqrt{h/kb}\,s/2 = 1.5$, the value of the quantity $(\tanh(\sqrt{h/kb}\,s/2)$ is approximately 0.9, indicating that 90% as much heat will be transferred in the area between the tubes as could be transferred with "infinite" spacing so that any increase in spacing would have a very small effect on the heat transferred by a given tube length. In those cases where tube length is a significant factor, this value would be an important criteria. In most cases, the total area is more important than tube length. A very significant factor is then the value of $\tanh(\sqrt{h/kb}\,s/2)/(\sqrt{h/kb}\,s/2)$ which is the ratio of the heat transferred by the fin portion, i.e., the portion between the tubes, to that which would be transferred if the fin area were at the constant temperature (T_W). This fin effectiveness factor is plotted versus tube spacing for a broad range of values of h/kb in figure 34. The fin effectiveness is a measure of the efficiency of the utilization for heat transfer of the area between the tubes.

The total heat which can be dissipated is the sum of the heat transferred by convection from the surface (the fin portion as given by equation (80) and the surface in a direct path from the tube) and the heat transferred by radiation

$$Q = h\ell\left[s_d + 2\sqrt{\frac{kb}{h}}\,\tanh\sqrt{\frac{h}{kb}}\,\frac{s}{2}\right](T_W - T_r) + (s_d + s)\,F_e F_a\,(T^4 - T_r^4) \qquad (81)$$

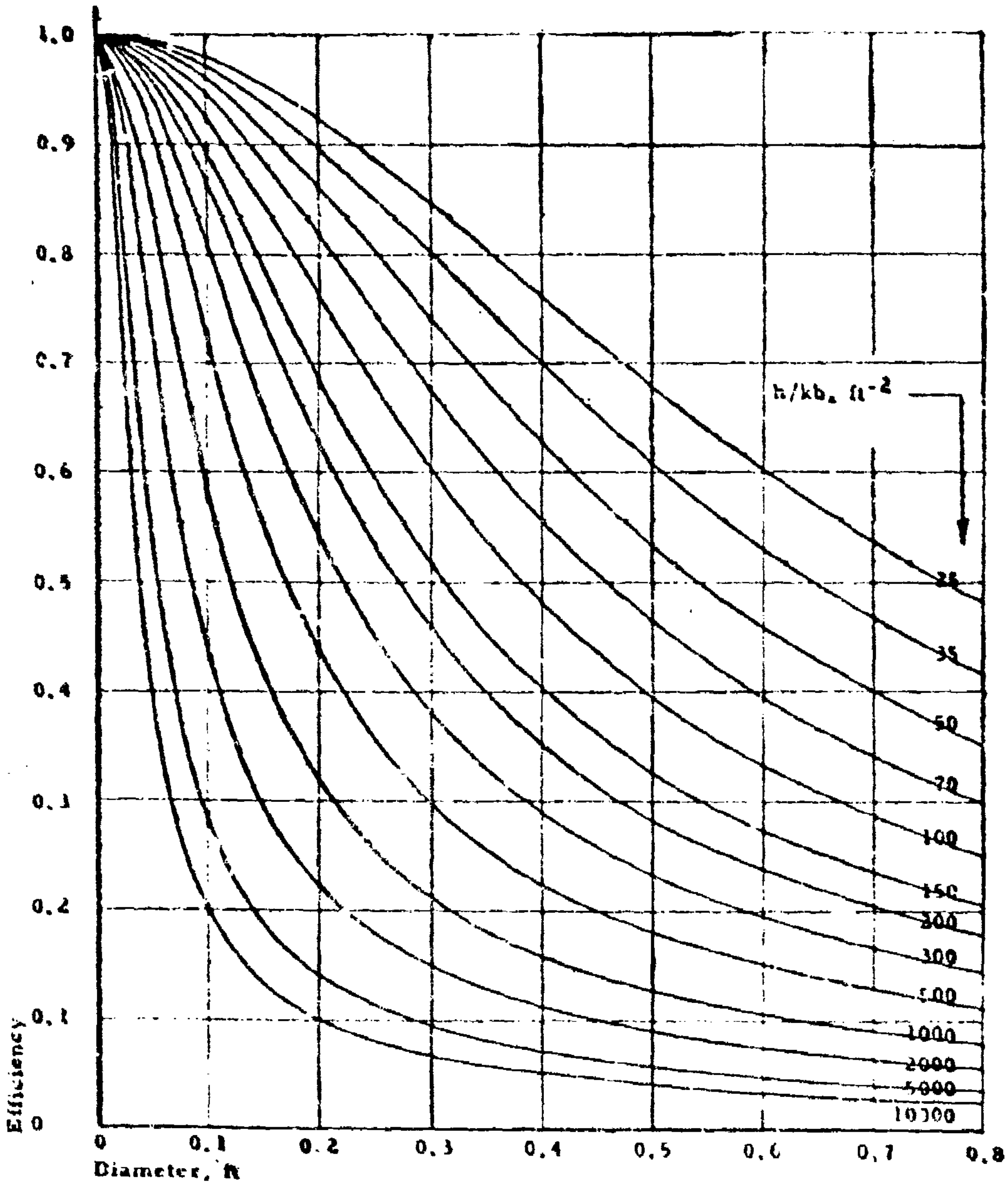


FIGURE 14 FIN EFFECTIVENESS VERSUS TUBE SPACING FOR A SURFACE CONDENSER

The effect of $\sqrt{h/kb}$ and $a/2$ is illustrated by figure 34. The curves show that the surface between the tubes is most effective in those cases where the value of $\sqrt{h/kb}$ is relatively small. If the tube spacing (a) is chosen to secure a given effectiveness, the heat transfer from the portion between the tubes is proportional to $h\sqrt{kb/h}$ or $\sqrt{hkb}$. The actual tube spacing should be selected considering the relative significance of total area and of tube length. For a given fin effectiveness and for a given percentage of maximum heat transfer per unit tube length, the tube spacing will vary directly with $\sqrt{h/kb}$.

The area that is required for a surface condenser is dependent on the following factors:

1) Required heat dissipation

2) External heat transfer coefficient

3) Difference between the surface temperature and the recovery temperature

The condenser must transfer (to the air) the heat produced by compression of the vapor and by compressor and motor losses in addition to the cooling load. The total heat that must be transferred by the condenser is then

$$Q_K = Q_V \left(1 + \frac{PI}{\eta_C \ \eta_M}\right) \tag{82}$$

The external heat transfer coefficient is primarily a function of the altitude and the flight velocity. Values of the heat transfer coefficient that are applicable in this study are plotted in figures 4 and 5.

The temperature of the condenser surface depends on the position relative to the condensing refrigerant. It is assumed the temperature drop through the wall is small so that a point directly through the tube wall from the condensing refrigerant is at a temperature T_K. The space between tubes is essentially a fin area and is treated, as indicated in the analysis above, in terms of a fin efficiency (η).

The required surface area for a given heat condenser load Q_K can be expressed in terms of the heat load (Q_K), the overall heat transfer coefficient (U), the direct heat flow path (s_d), the tube spacing (s), and (assuming a constant condensing temperature the difference between the condensing temperature (T_K) and the recovery temperature (T_r). The area is

$$A_S = \frac{Q_K (s_d + s)}{U (s_d + \eta s) (T_K - T_r)} \quad (83)$$

The weight is equal to the area times the weight per unit area, then, using a factor (F_S) for the effect of fin efficiency on areas and the ratio of the outside to the overall heat transfer coefficient,

$$F_S = \frac{h_o (s_d + s)}{U (s_d + \eta s)} \quad (84)$$

A factor which is quite near unity for nearly all cases

$$1 < F_S < 1.1$$

The weight of the surface heat exchanger is then

$$W_S = \frac{F_S g \rho_b Q_K}{h_o (T_K - T_r)} \quad (85)$$

D. Results of Vapor Cycle Cooling System Analysis

Vapor cycle cooling systems of the type shown in figure 8 have been analyzed for the entire altitude-Mach number range of interest in this study. Vapor cycle systems in combination with expendable coolant systems have also been analyzed. Such systems are discussed in section VIII of this report. Variations of the basic vapor cycle system as illustrated in figures 9, 35, and 36 have been analyzed for certain conditions to determine the relative effect of the variations. Such variations are discussed following the general results.

The systems have been evaluated in terms of a total equivalent weight as defined in section IV of this report.

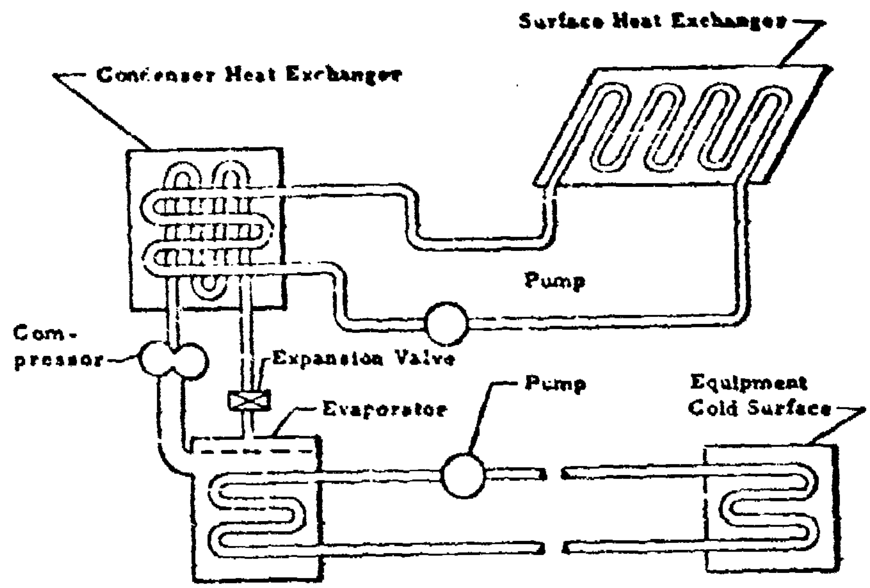


FIGURE 35 VAPOR CYCLE COOLING SYSTEM WITH LIQUID COOLED CONDENSER

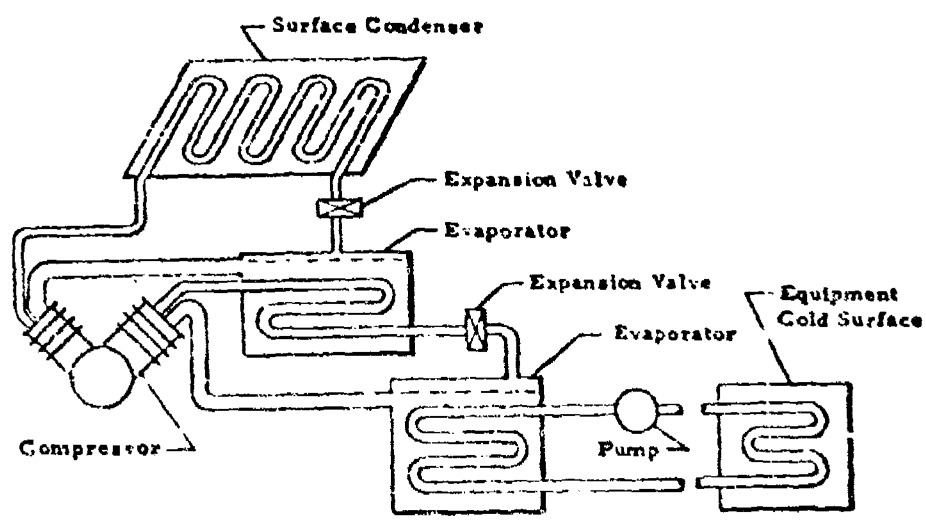


FIGURE 36 CASCADE VAPOR CYCLE COOLING SYSTEM

The assumed operating conditions include the altitude and Mach number range of figure 1 and equipment exit temperatures of 160 °F and 275 °F. The equipment cooling load is assumed to be 10 kw. The results are considered typical for units somewhat smaller as well as for units several times as large. The larger units would permit a slight weight reduction on a per kilowatt basis.

The most significant fundamental effects of vapor cycle cooling systems are the weight addition and the power requirement. Systems utilizing a condenser cooled by ram air would impose a significant drag. For systems employing a surface condenser or a heat exchanger, the small increase in drag will be neglected as discussed in section V of this report.

The factors that vary between different velocities and altitudes are the weights of the compressor, evaporator, and condenser and the power input to the compressor. The weights and power associated with the heat transport fluid and with the absorption of the heat at the equipment is essentially independent of the cooling system for the types of systems under consideration. Any variations would be caused by differing significance being assigned to various factors in the optimization procedure; however, because of the basic similarity of various systems, a heat transport system that is optimized for one system will be very near the optimum for other vapor cycle systems. Consequently, the weight of the heat transport system is not included in the curves defining the effects of altitude and velocity and illustrating the differences and the range of applicability of Freon-11 and water systems. The figures are drawn assuming an optimized heat transport system in so far as it will affect the evaporator temperature and weight, the compressor and power supply system weight, and the power input. The total equivalent weights as given include the weights of the compressor, the power supply system, the evaporator, the condenser, and a factor of one (or two) as an equivalent weight factor for the total power required. An additional factor of one times the power requirement is included to account for the connections, lines, and the expansion valve. The weight of such items depends on size and strength requirements which in turn are determined by the pressure considerations and volume flow requirements. These latter factors are related to the power requirement. The weight of the heat transport system is considered equal for the different systems of this type. The weight of the transport system, dependent on the distribution of equipment, can be determined by reference to section IV. For a 10 kw cooling load and assuming an average or effective distance of twenty feet, an equivalent weight of ten pounds may be considered typical.

Equivalent weight is plotted versus flight Mach number for various altitudes and equipment exit temperatures in figures 37 to 40. Curves are drawn using power translation factors $f_{p\ eq}$ of one and two as discussed in section III. The effect of other translation factors can be readily determined by noting the effect as indicated by the figures. The difference in the Mach number as given on the abscissa scale for Freon-11 and water should be noted. Also it should be remembered that the curves for Freon-11 do not reflect any fluid instability which may preclude use at the higher Mach numbers. The figures indicate that for similar conditions below Mach 2, the total equivalent weights of Freon-11 and water systems are quite comparable; as the velocity increases above Mach 2, the total equivalent weight of the Freon-11 system increases much more rapidly than does the weight of the water system. The maximum practical operating limit for Freon-11 is about Mach 2.3 at the higher equipment temperatures and under 2.2 for the lower equipment temperatures. Systems using water as the refrigerant will operate with reasonable total equivalent weights, considering the temperature level to which the heat must be raised, for the entire altitude-Mach number range being considered in this study. This is primarily a result of the high critical temperature of water (705°F) and of the high latent heat of water which is several times as large as for Freon-11.

The effect of altitude on the total equivalent weight is illustrated by figures 41 to 44 for Freon-11 and for water. The minimum weight at 40,000 feet altitude is a reflection of the lower recovery temperature at that altitude and of the relatively high heat transfer coefficient that applies. The limits in the altitude-Mach number envelope for which systems of various total equivalent weights will provide cooling at the specified equipment exit temperatures are shown in figures 45 and 46 for Freon-11 and in figures 47 and 48 for water. The systems have been optimized for a surface heat transfer coefficient (h_o = 20). These curves indicate that a system of a given weight will be capable of cooling equipment, maintaining the specified exit temperature, for any flight condition within the specified envelope, to the left of the respective curve. Optimization at h_o = 20 (altitude approximately 55,000 feet) implies that operation at other conditions is somewhat off design. That is, for high altitudes for which the external heat transfer coefficient is lower, a somewhat larger surface area would result in less total equivalent weight. In the assumed configuration, the drop in heat transfer coefficient must of course be compensated for by an increase in temperature difference (T_r - T_K).

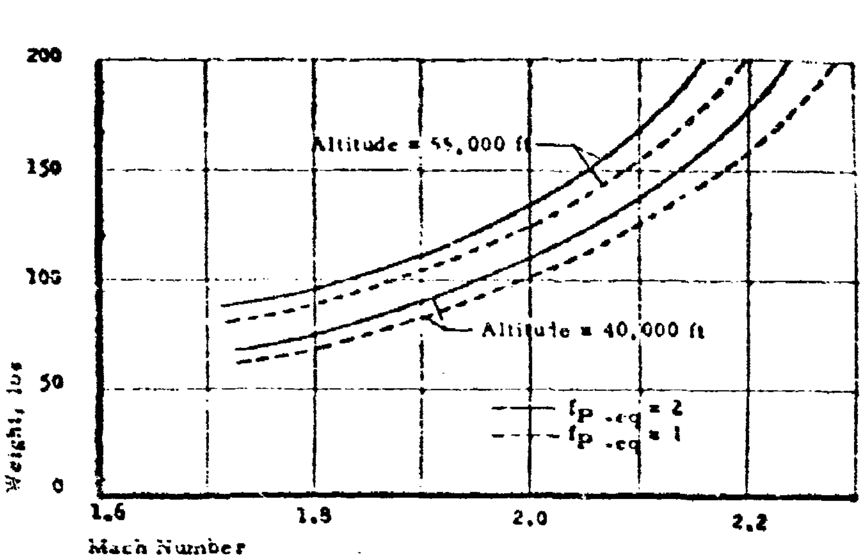


FIGURE 37 TOTAL EQUIVALENT WEIGHT VERSUS MACH NUMBER FOR A FREON-11 VAPOR CYCLE COOLING SYSTEM (T_{Ee} = 160°F)

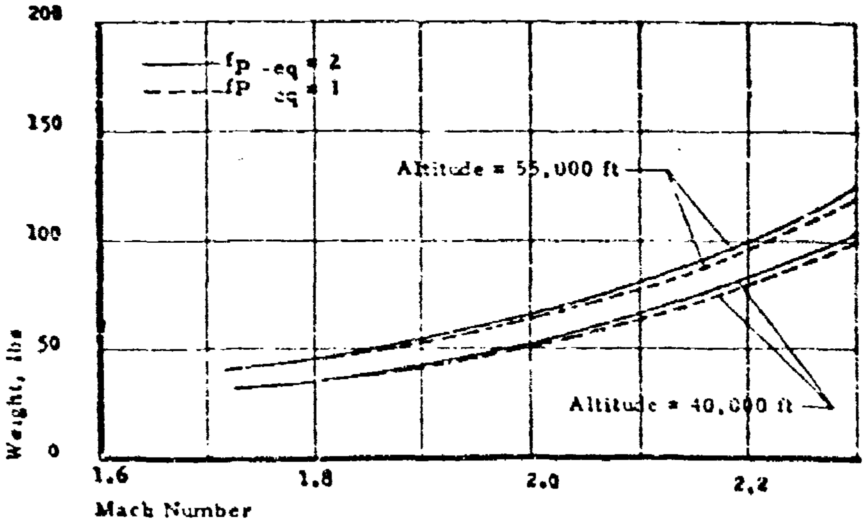


FIGURE 38 TOTAL EQUIVALENT WEIGHT VERSUS MACH NUMBER FOR A FREON-11 VAPOR CYCLE COOLING SYSTEM (T_{Ee} = 275°F)

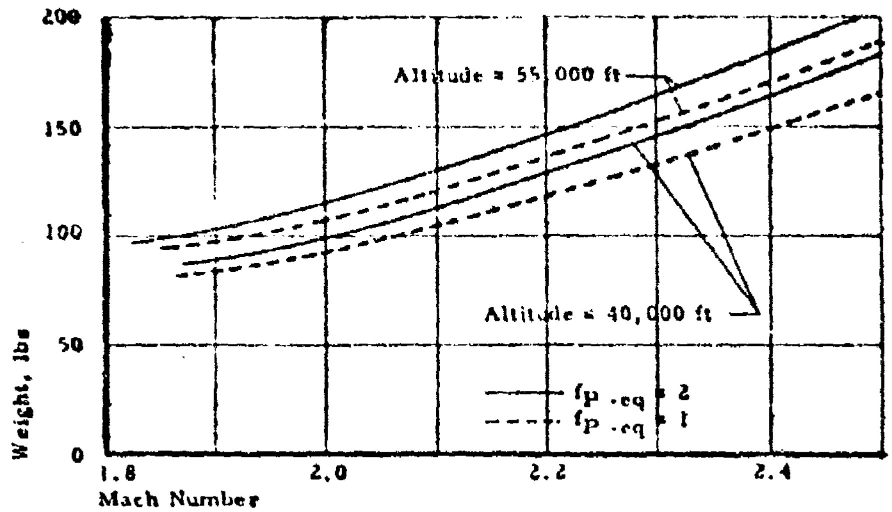


FIGURE 39 TOTAL EQUIVALENT WEIGHT VERSUS MACH NUMBER FOR A WATER VAPOR CYCLE COOLING SYSTEM (T_{Ee} = 260°F)

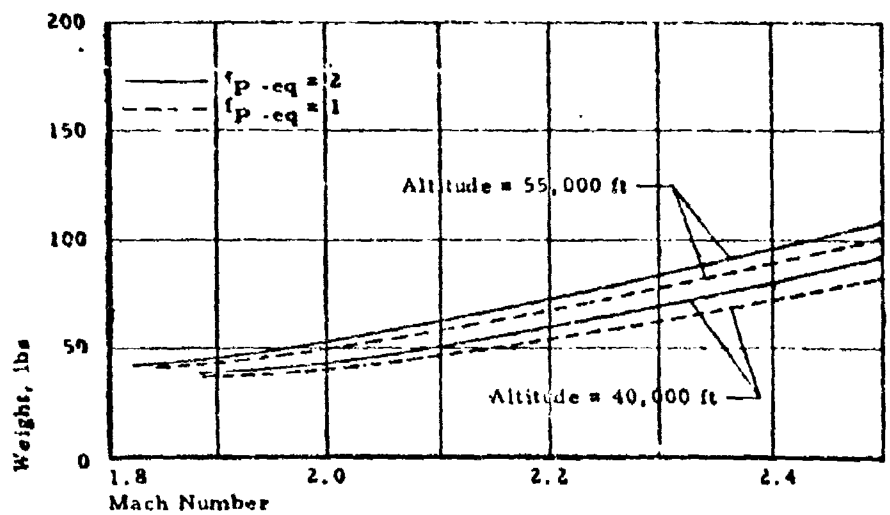


FIGURE 40 TOTAL EQUIVALENT WEIGHT VERSUS MACH NUMBER FOR A WATER VAPOR CYCLE COOLING SYSTEM (T_{Ee} = 275°F)

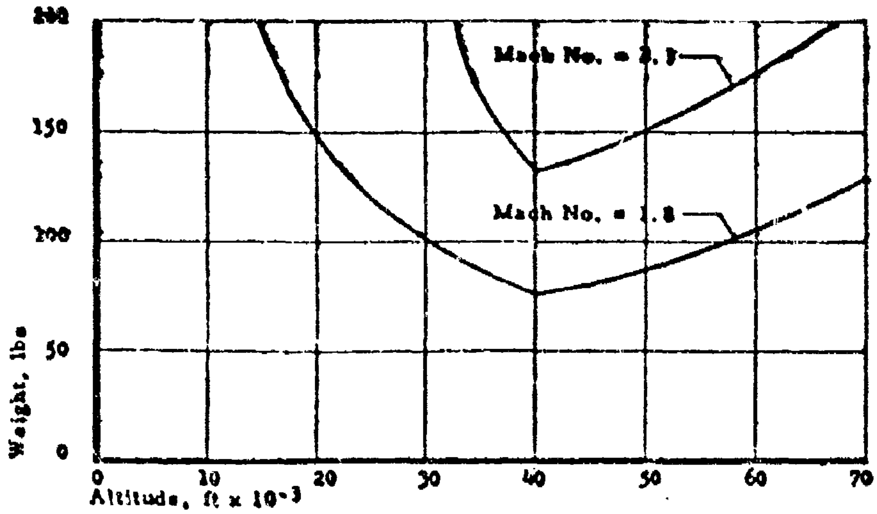


FIGURE 41 TOTAL EQUIVALENT WEIGHT VERSUS ALTITUDE FOR A FREON-11 VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 160°F)

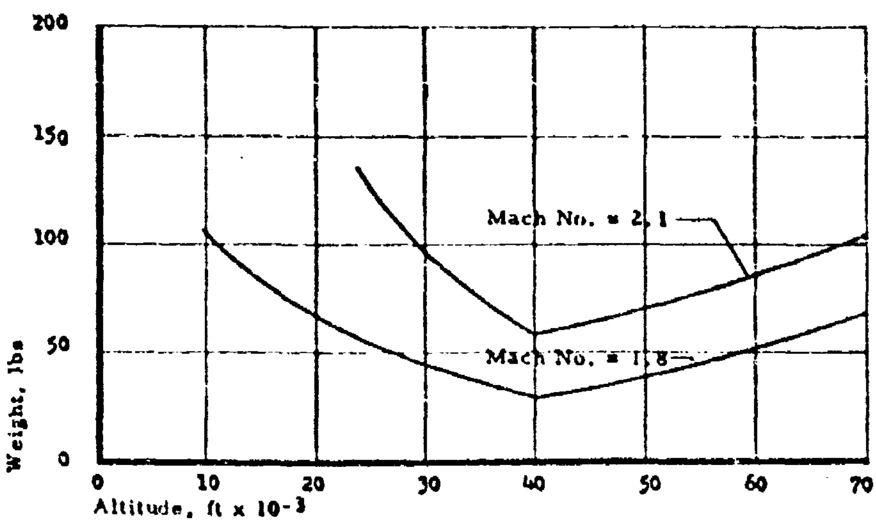


FIGURE 42 TOTAL EQUIVALENT WEIGHT VERSUS ALTITUDE FOR A FREON-11 VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 275°F)

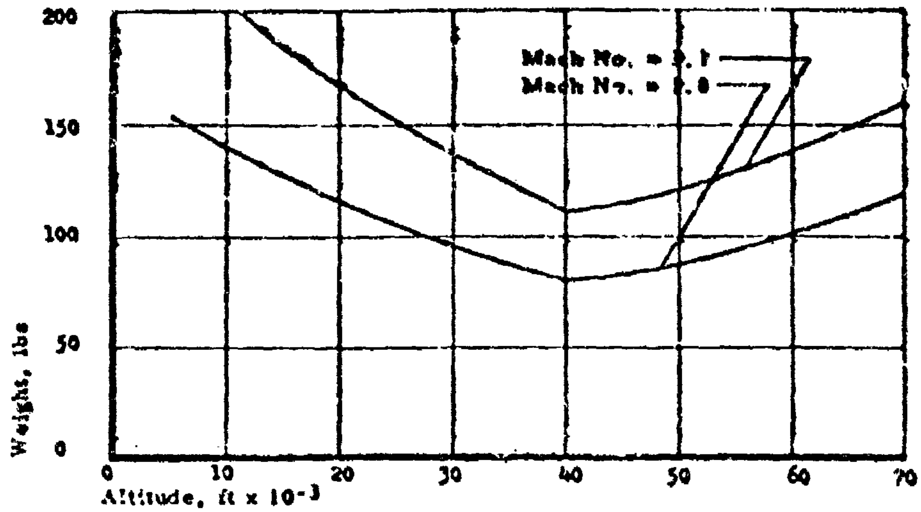


FIGURE 43 TOTAL EQUIVALENT WEIGHT VERSUS ALTITUDE FOR A WATER VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 160°F)

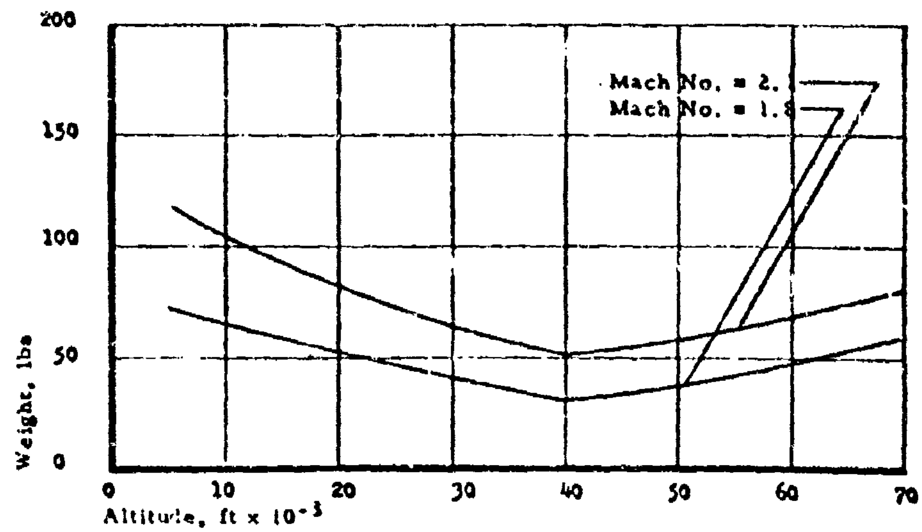


FIGURE 44 TOTAL EQUIVALENT WEIGHT VERSUS ALTITUDE FOR A WATER VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 275°F)

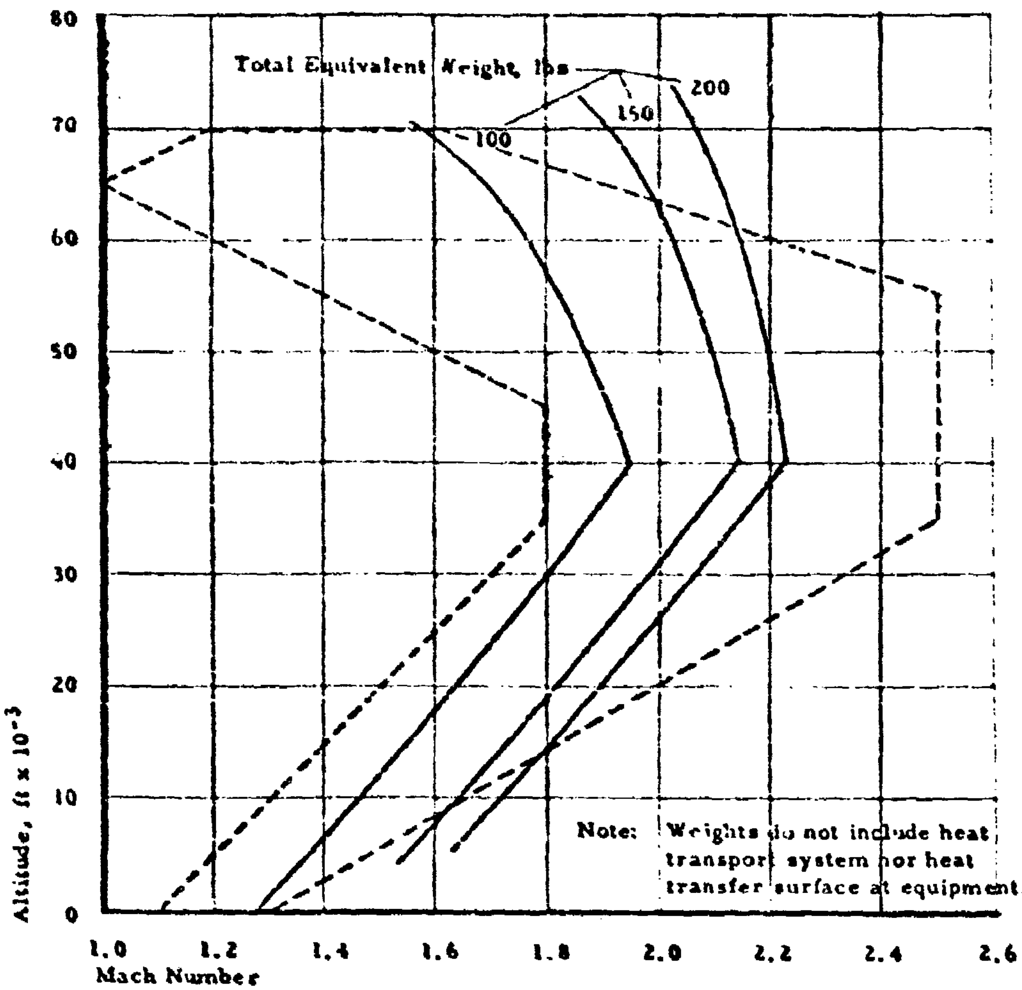


FIGURE 45 COOLING SYSTEM OPERATING LIMITS FOR GIVEN TOTAL EQUIVALENT WEIGHTS FOR A FREON-11 VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 160°F)

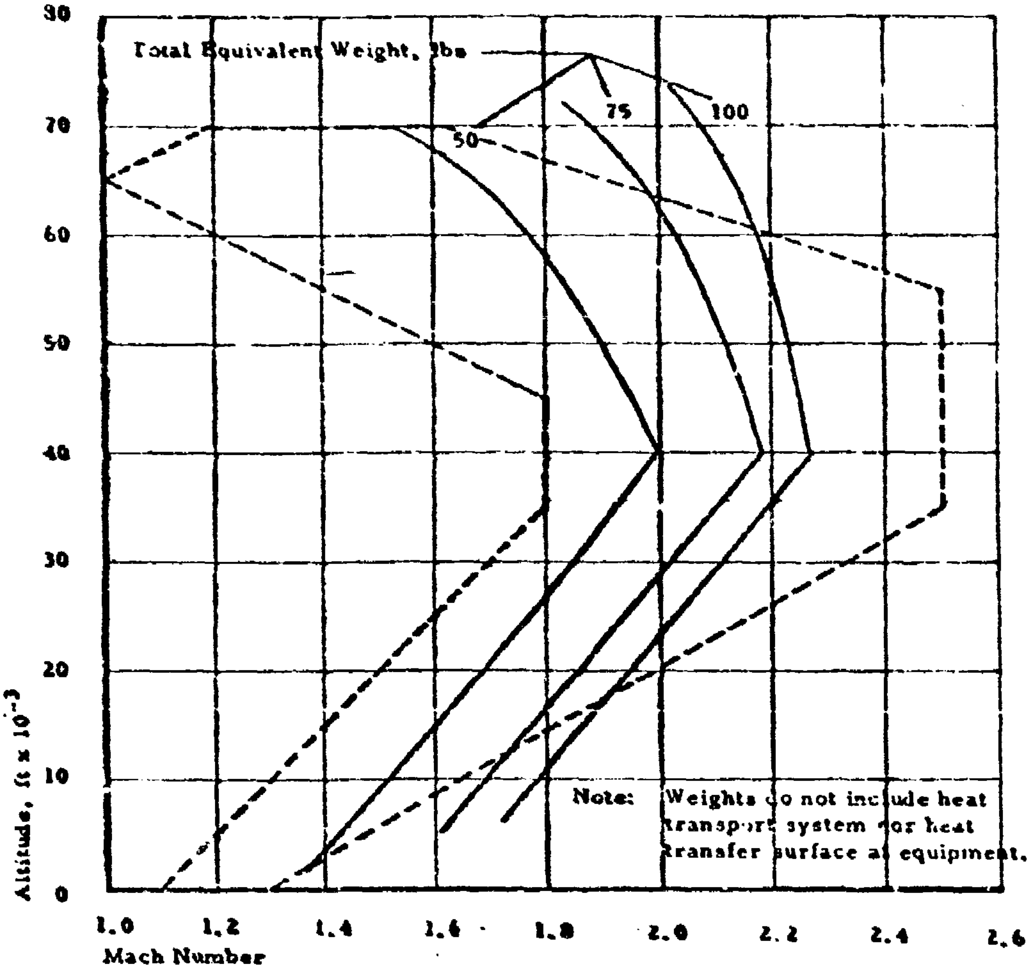


FIGURE 46 COOLING SYSTEM OPERATING LIMIT FOR GIVEN TOTAL EQUIVALENT WEIGHTS FOR A FREON-11 VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 275°F)

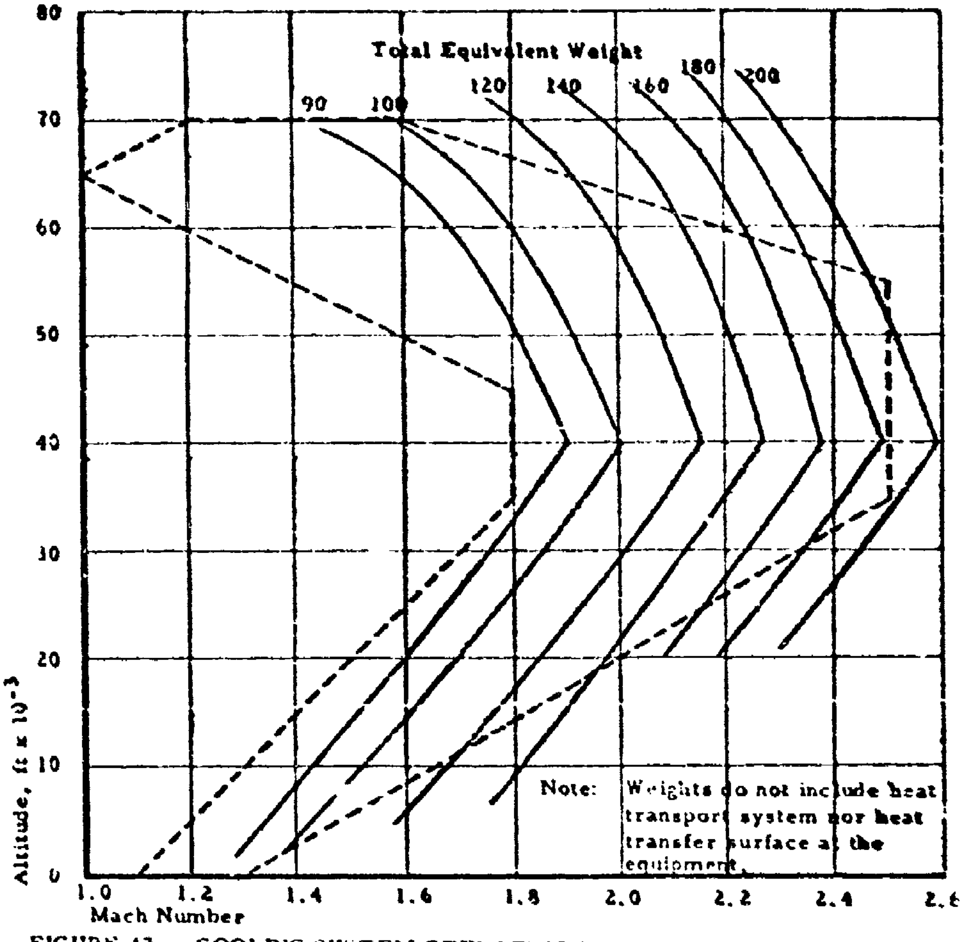


FIGURE 47 COOLING SYSTEM OPERATING LIMITS FOR GIVEN TOTAL EQUIVALENT WEIGHTS FOR A WATER VAPOR CYCLE COOLING SYSTEM (T_{Ee} = 160°F)

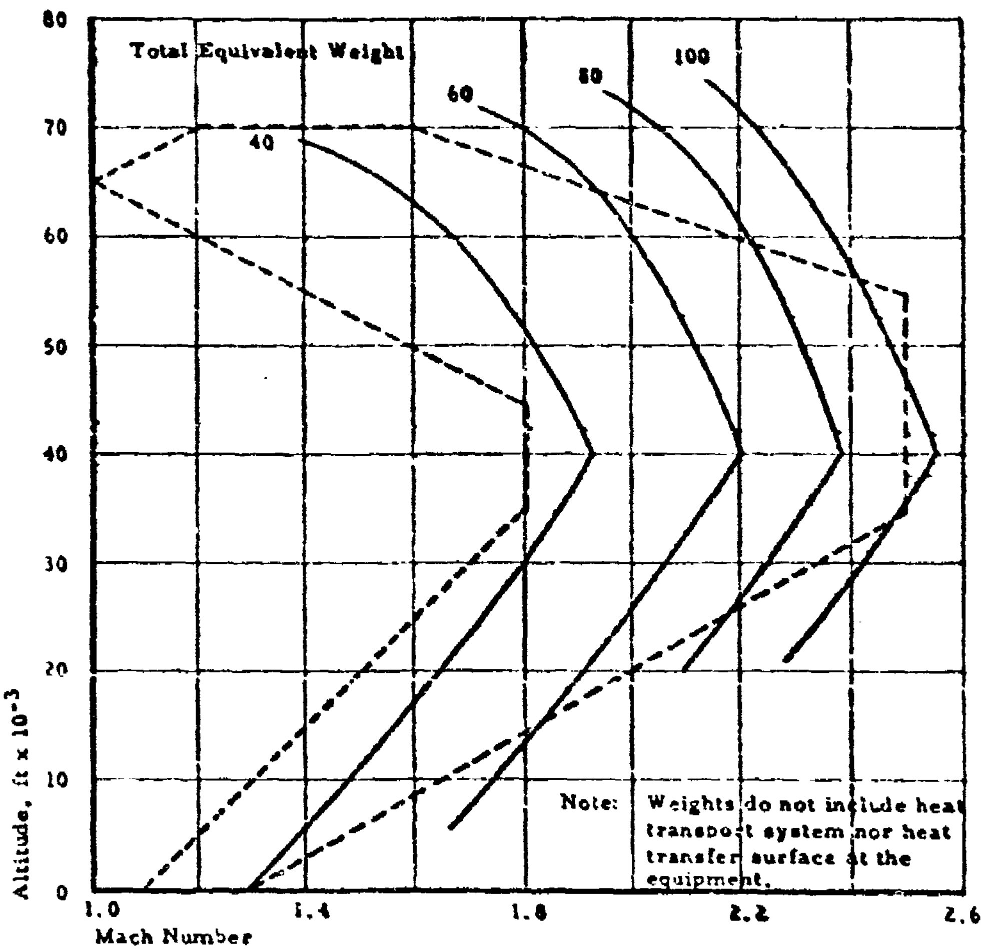


FIGURE 48 COOLING SYSTEM OPERATING LIMITS FOR GIVEN TOTAL EQUIVALENT WEIGHTS FOR A WATER VAPOR CYCLE COOLING SYSTEM (T_{E_e} = 275°F)

The effect of optimizing at various surface heat transfer coefficients (essentially at different altitudes) is indicated by the curves of figure 49. All curves are for equal total equivalent weight. Each curve is for a somewhat different system design in so far as condenser area and temperature difference is concerned. The weight line at the right (dashed line) represents the maximum velocity for which a system of the given total equivalent weight can cool the equipment when designed for optimum operation at each altitude. The system, therefore, involves no off-design operation but does not represent a single system design.

Vapor cycle systems are particularly attractive as unlimited-duration cooling systems for equipment in which the heat can be readily transferred to a cold plate either by conduction, vaporization, or convection. The systems do not lend themselves readily to the concept of individualized equipment cooling.

E. Effect of Variations from the Assumed Configuration

The general analysis of vapor cycle systems was made assuming a high speed lightweight positive displacement rotary compressor of the type illustrated in figure 50 (assuming compressor weight as given by equation 54). However, the results are quite representative even for somewhat heavier but well designed reciprocating compressors illustrated schematically in figure 51 (assuming the compressor weight is given by equation 53). This anomaly is caused by the fact that the reciprocating compressors are usually somewhat more efficient than the rotary type, as is pointed out in the discussion of compressors in section V-C of this report. A more efficient compressor results in a lower power requirement and in less heat having to be dissipated by the condenser. The weights of various components and the horsepower input times a factor of three, for a 10 kw cooling load and an equipment exit temperature of 212°F are shown in figure 52 assuming a positive displacement rotary compressor with a total efficiency of 65% and in figure 53 assuming a well-designed reciprocating compressor with a total efficiency of 80%.

A modification of the basic vapor cycle cooling system is illustrated in figure 35. This system utilizes an intermediate liquid-cooled condenser. A surface heat exchanger is used to remove the heat from the condenser cooling liquid. The system was conceived to eliminate the need for a high pressure surface condenser and the long high pressure lines to the surface condenser. At the higher temperatures, the vapor pressure of water is quite high (680 psi at 500°F). With this modification of the basic vapor cycle cooling system, the condenser would be a compact liquid cooled unit located very near the compressor, thus eliminating the need for long high

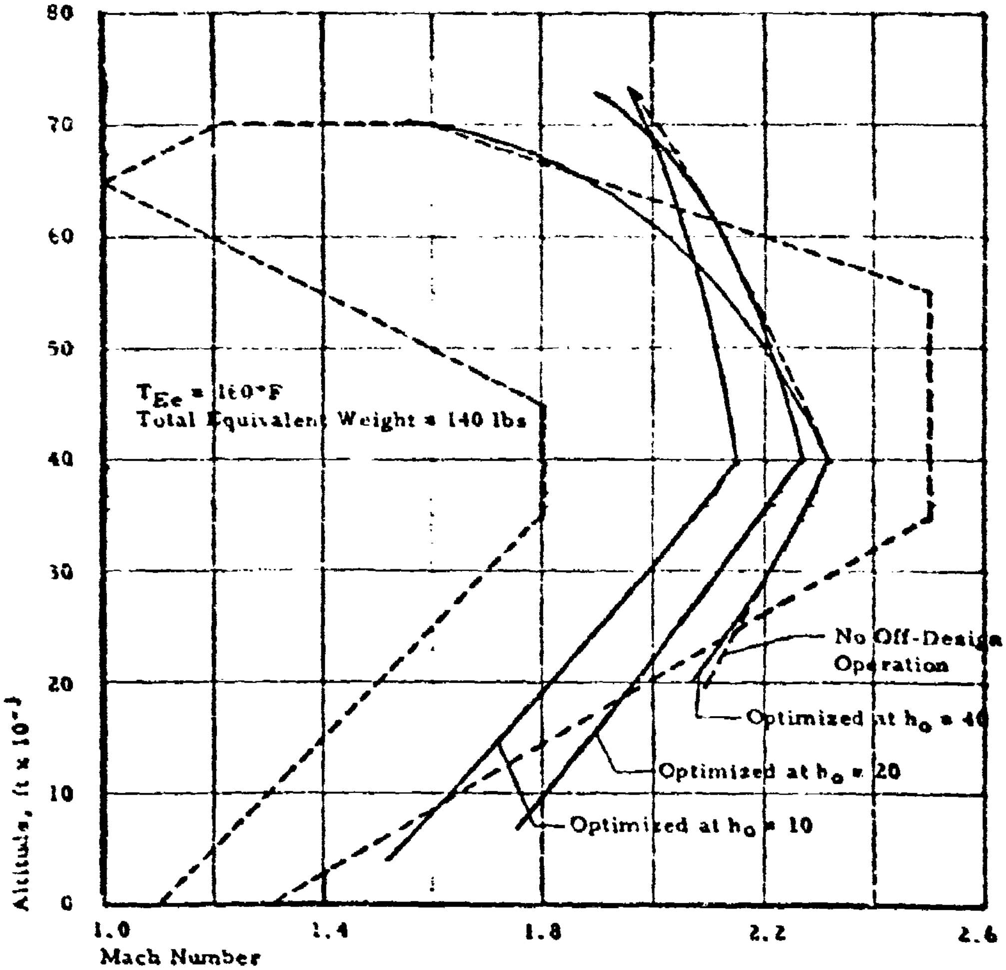
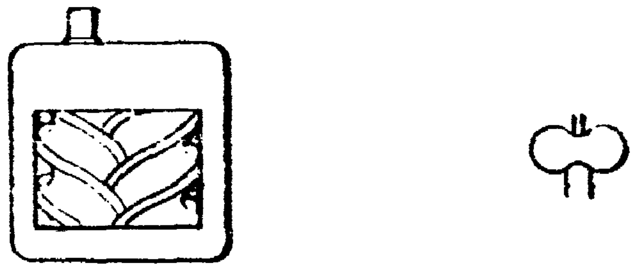
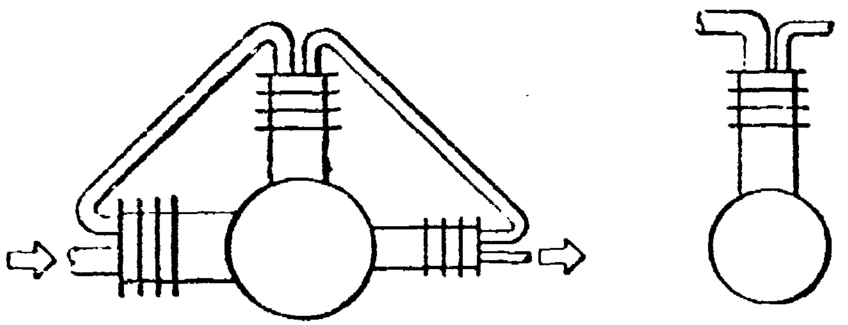


FIGURE 49 EFFECT OF OFF-DESIGN OPERATION OF A WATER VAPOR CYCLE COOLING SYSTEM



a. Typical Unit b. Schematic

FIGURE 50 POSITIVE DISPLACEMENT HELICAL TYPE COMPRESSOR



a. Typical Unit b. Schematic

FIGURE 51 THREE STAGE RECIPROCATING TYPE COMPRESSOR

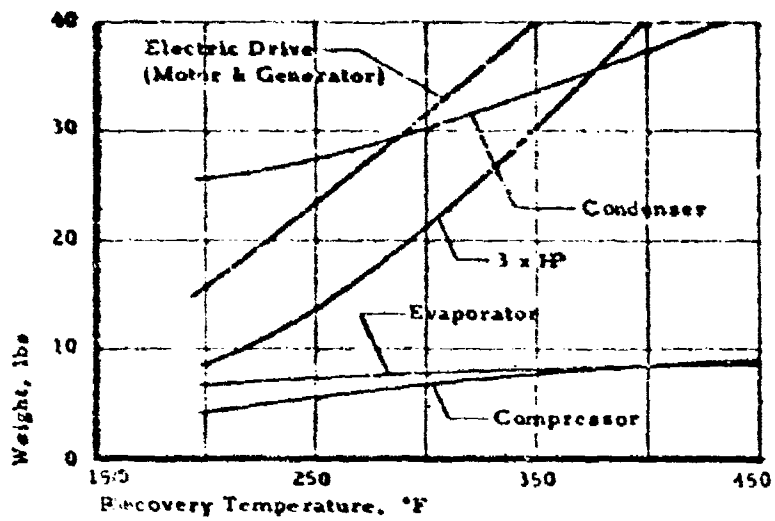


FIGURE 52 WEIGHT OF INDIVIDUAL COMPONENTS OF A WATER VAPOR CYCLE COOLING SYSTEM USING A HELICAL TYPE COMPRESSOR (T_{E_e} = 210°F)

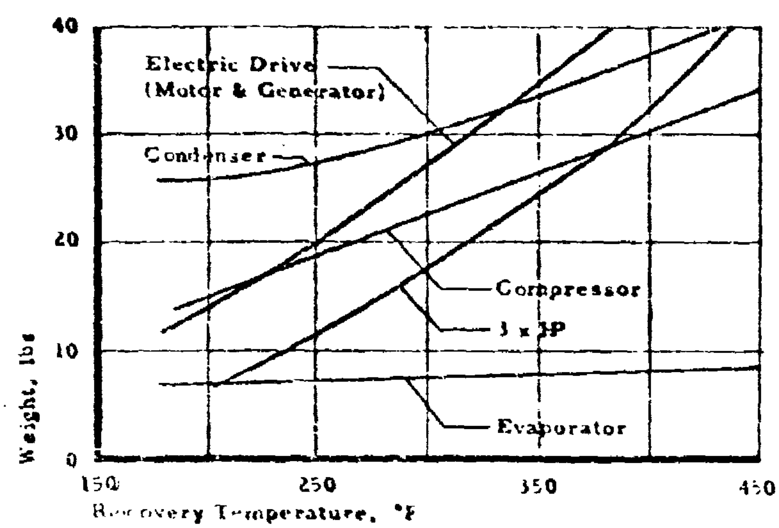


FIGURE 53 WEIGHT OF INDIVIDUAL COMPONENTS OF A WATER VAPOR CYCLE COOLING SYSTEM USING A RECIPROCATING TYPE COMPRESSOR (T_{E_e} = 210°F)

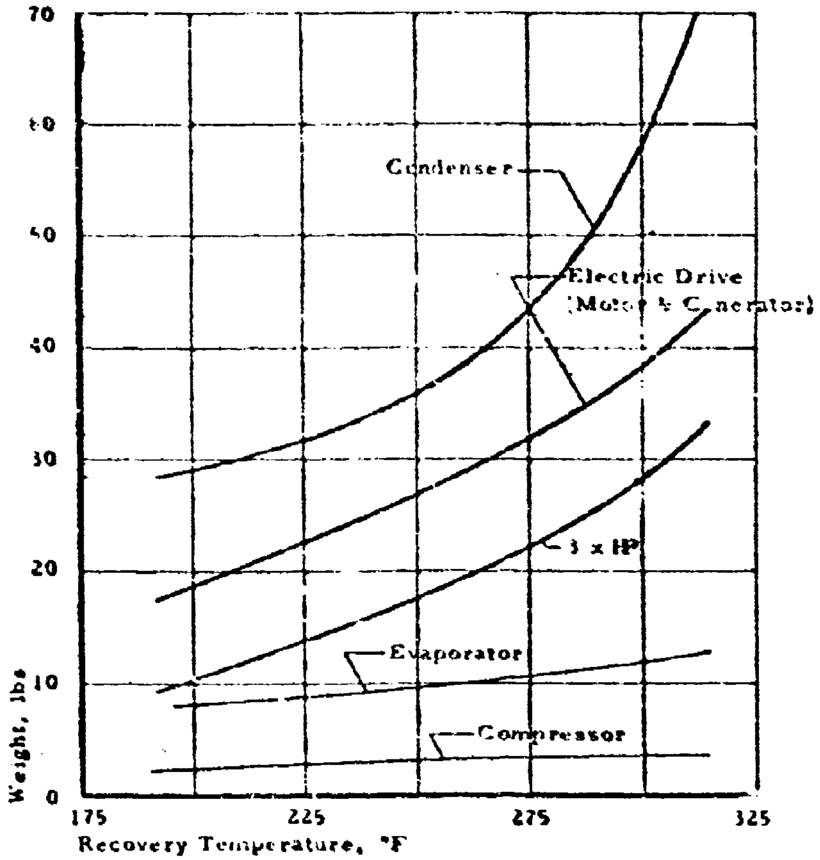


FIGURE 54 WEIGHT OF INDIVIDUAL COMPONENTS OF A FREON-11 VAPOR CYCLE COOLING SYSTEM USING A HELICAL TYPE COMPRESSOR (T_{E_e} = 210°F)

pressure lines. The liquid used to transport the heat to the surface heat exchanger would be one with a very low vapor pressure at the higher temperatures, such as Dowtherm A which has a vapor pressure of about 15 psia at 500°F. This modification of the basic vapor cycle cooling system entails an additional heat exchanger which imposes an additional weight and also a temperature drop. The temperature drop in the condenser liquid heat exchanger can be kept small because of the high heat transfer coefficients that apply. For the same reason, the size and weight of the unit are relatively small. The unit for a typical system would weigh on the order of ten pounds. A part of this weight could be saved through the use of a lighter surface heat exchanger which is made possible by the lower pressures. The overall system total equivalent weight would therefore increase by 5% or less. This modification is considered practical for the higher temperature water vapor cycle applications.

A vapor cycle cooling system similar to the type assumed in the general study (figure 8) but with a ram air cooled condenser instead of the surface condenser is illustrated in figure 9. This cooling system is very similar to the other systems in so far as the heat absorption, compression, expansion, and controls are concerned. The basic differences in the condenser result in somewhat different characteristics that require an analysis to determine the difference in the effects. The major differences are:

1) The system requires ram air for condenser cooling. This requirement entails the use of air from the engine inlet or a separate inlet diffuser to take the ram air on board, ducts to convey the air to and from the condenser, and an exhaust nozzle for thrust recovery.

2) The system transfers heat to air taken on board the aircraft which is initially at the total ram temperature. The air temperature increases as the heat from the condenser is absorbed. The condensing temperature must be above the final air temperature.

3) The system will impose a significant momentum drag due to taking ram air on board the aircraft.

The use of ram air for cooling and the drag effects incurred by the use of ram air are analyzed in section VI of this report. Only the directly applicable equations will be presented here. The amount of air that must be taken on board to cool the condenser is dependent on the cooling load, the power input, the compressor and motor efficiencies, the condenser effectiveness, and the condenser ram total temperature.

The ram airflow can be expressed by the equation

$$w_a = \frac{3.95\ kw}{c_K\,(T_K - T_T)}\left(1 + \frac{PI}{\eta_C\,\eta_M}\right) \tag{86}$$

The net momentum drag that will be imposed on the aircraft is the drag associated with taking the air on board minus the thrust recovered when the air is exhausted. That drag is given by the equation

$$D_{mom} = w_b\,(34.7\sqrt{\theta_a}\,M - F_{th}\sqrt{\theta_{ne}}) \tag{87}$$

For supercritical flow

$$\frac{p_a}{p_{ne}} < 0.528,\quad F_{th} = 54.4\sqrt{\theta_{ne}}\ \eta_n\left(1 - \frac{p_a}{p_{ne}}\right)$$

For subcritical flow

$$\frac{p_a}{p_{ne}} \geq 0.528,\quad F_{th} = 78.8\sqrt{\theta_{ne}}\ \eta_n\left[1 - \left(\frac{p_a}{p_{ne}}\right)^{0.286}\right]^{1/2}$$

With a vapor cycle system, there is no blower or compressor in the ram air ducts and consequently the amount of thrust that can be recovered is limited by the inlet pressure ratio. The pressure drops through the condenser and ducts reduce the available nozzle pressure ratio. The duct inlet pressure ratios versus Mach number are plotted in figure 64 assuming a supersonic diffuser with 85% recovery. The weights of the ram air cooled condenser can be determined by reference to figure 63 in which weights of boiler heat exchangers are given in terms of pressure drop, effectiveness, and airflow rates. The weights of a ram cooled condenser would be comparable provided that the condensing pressures are not excessive.

The momentum drag and condenser weight will decrease as the ram air flow decreases; however, the condensing temperature must then be increased to dissipate the heat which in turn results in increased power input and a higher equivalent weight for the balance of the system. Therefore, a compromise must be reached to obtain an optimum cooling system.

The use of a ram air cooled condenser results in a system with a lower maximum operating limit in so far as Mach number is concerned and a considerably greater total equivalent weight for equivalent flight conditions. The increase in total equivalent weight for a cooling system with a ram air condenser as compared to one with surface condenser is typically from 25 to 30% depending on the operating conditions. The increase is primarily a result of (1) the increase in the power input due to the increased condenser temperature, (2) the momentum drag of ram air, and (3) the weights of air ducts.

The advantage of this type of system is that it does not require the rather large aircraft surfaces which must be available for the surface condenser. It should further be pointed out that in the event that a surface condenser caused premature transition to turbulence, a significant drag would be imposed by the surface condenser which has not been included in the present study.

A cascaded vapor cycle cooling system is illustrated in figure 36. This system is different from the basic system being considered in that the heat is pumped to the final sink temperature in two steps. The system actually involves two complete vapor cooling cycles with the evaporator of the high temperature stage serving as the heat sink for the low temperature stage. In the illustration, a single vapor compressor is indicated with the low pressure stage being used for the low temperature cycle and the second compressor stage being used for the high temperature cycle. In this version, the same refrigerant would be used in the entire system. The concept could also be used with separate compressors and different refrigerants.

This type of cooling system involves a temperature drop (relatively small because of the high heat transfer coefficeints) in the interstage condenser-evaporator unit. Because of this temperature drop, the system should not be considered unless the overall temperature difference is quite large (about 100°F or more). The system does involve considerable additional mechanical complexity and is therefore considered practical only for applications that involve high power inputs. The total power input for a cascade cycle is

$$PI_T = PI_1 + PI_2 (1 + PI_1) \qquad (88)$$

The condenser temperature of the first stage must be above the evaporator temperature of the second stage. A definite advantage in terms of the power input will be secured for cycles using refrigerants that are sensitive to the difference between the evaporator and the condenser temperature. Thus the Freon-11 systems can be significantly improved by means of a cascade cycle, while the water vapor cycle systems will show only a slight improvement. Basically any vapor cooling cycle operating between temperatures for which the power input is reasonably near the Carnot value will show little improvement. Improvement in the cycle power input is limited to a reduction in the deviation from the Carnot cycle power input. The Carnot power input is

$$(PI)_{Carnot} = \frac{T_K - T_V}{T_V} \qquad (89)$$

A Freon-11 cooling system with an evaporator temperature of 150°F and a condenser temperature of 350°F would require a power input of 0.87 for a simple cycle. A cascade cycle with the first stage working from 150° to 260°F and the second stage from 240° to 350°F would have a power input of 0.59, a power saving of about 30%. It is interesting to note that a simple water cycle for the same conditions (150° to 350°F) would have a power input of 0.46 or about 20% less than the cascaded Freon-11 cycle and 47% less than the simple Freon-11 cycle. The water cooling cycle cascaded in the same manner as the Freon-11 cycle would have a power input of 0.44, a power saving of only about 4% as compared with the simple water cycle. The above examples bear out the previous observation that the cascade cycle can only bring the power input nearer the Carnot value which in this case is 0.328.

The absorption vapor cycle is a variation of the basic vapor cooling cycle quite different in operation from the compressor vapor cycle. An absorption vapor cycle cooling system is shown schematically in figure 55. Actual design would involve special problems, such as securing a stable unit and one that would operate satisfactorily in an aircraft application. The absorption vapor cycle has not been analyzed in detail in this study. In this cycle, the fluid in the vapor state after evaporation is dissolved by a solute at the evaporation pressure. The solution, rich in refrigerant (in the liquid state), is then pumped to a generator at the condenser pressure. The refrigerant is changed in state and driven out of solution by the addition of heat and enters the condenser in the vapor state. The solute is returned to the absorber after the refrigerant is driven off. The absorption cycle thus differs from the compression cycle by the

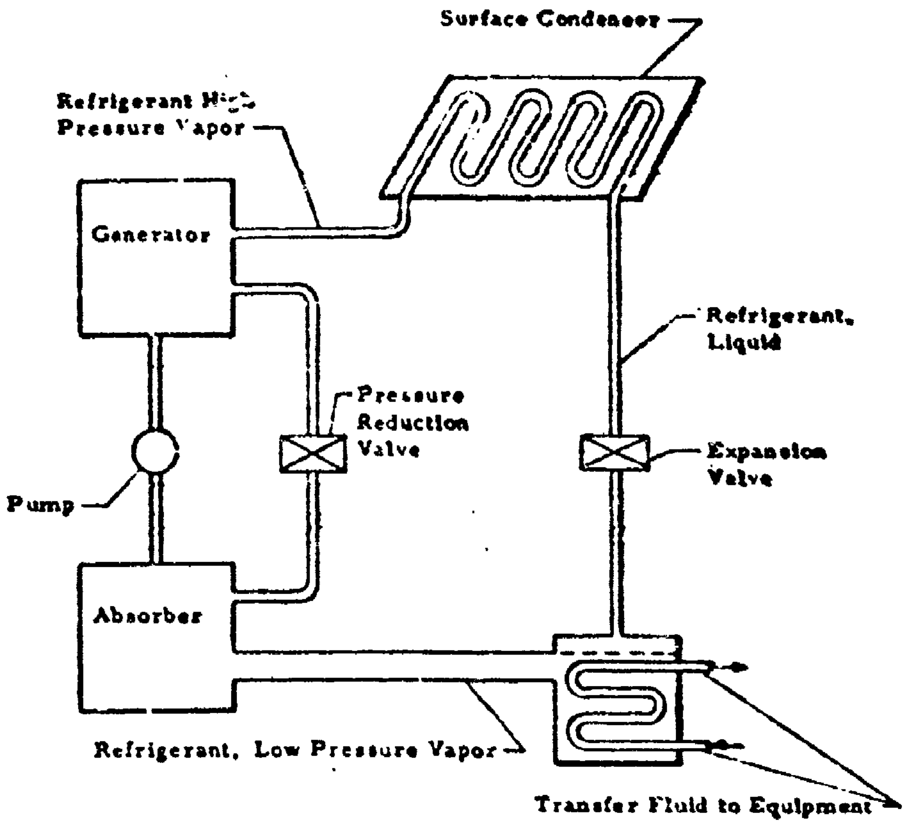


FIGURE 55 ABSORPTION VAPOR CYCLE COOLING SYSTEM

substitution of an absorber and generator for the compressor and by the use of energy in the form of heat instead of mechanical work. The cycle also differs in that there are additional changes of state from vapor to liquid in the absorber and from liquid to vapor in the generator. Means must be furnished to pump the liquid from the low (evaporator) pressure to the high (condenser) pressure in the absorption cycle. The heat energy for the absorption cycle must be available at a higher temperature than the operating condenser temperature. The fact that the vapor undergoes a change of state when going into solution indicates that the heat of vaporization must be added in the generator; consequently, the cooling effect cannot be greater than the energy input even for the ideal case. The PI will be greater than one. As pointed out in the previous discussion, the compression cycle, being a true pumping system, does not have this limitation. The PI for many cases is much less than one. An inherent advantage of the absorption cycle is the elimination of the relatively large displacement compressor. The absorption cycle may have an additional advantage if the necessary heat energy is more readily available or involves less adverse performance effects than would the smaller amount of mechanical energy necessary to operate a compression cycle. The practicality of an absorption cooling system would depend on the finding of a refrigerant and a solute with the necessary properties. In addition to the refrigerant properties as discussed in a previous section of this report, the refrigerant and solute must have certain special properties. Solubility and vapor pressure characteristics at the applicable temperatures would be a prime consideration. Among the possibilities would be water as the refrigerant with a brine solution as the solute. The weight of such a system would have to be carefully determined as this would likely be a major factor. This type of system is mentioned as a possibility but has not been further analyzed in this study.

F. Conclusions With Regard to Vapor Cycle Cooling Systems

1. Freon-11 Cooling Systems

The following conclusions can be drawn from the analysis of vapor cycle cooling systems using Freon-11.

1) The systems using Freon-11 as the refrigerant are limited to flight at a Mach number of about 2.2 and 2.0, for equipment exit temperatures of 275° and 160°F respectively, when a surface type condenser is used. The use of a ram air cooled condenser would result in a limiting Mach number of approximately 2.0 and 1.8 for the high and low equipment temperatures.

2) The total equivalent weights are a minimum for flight at an altitude of 40,000 feet increasing roughly fifty pounds for flight at equal Mach number and 70,000 feet. This is primarily due to the decrease in the heat transfer coefficient. The increase in the total equivalent weight at the lower altitudes, while much greater due to the higher temperatures, is probably not as significant since it is reasonable to assume that actual flight velocities would be at lower Mach numbers at the lower altitudes.

3) The total equivalent weight is a little more than twice as large for equipment exit temperatures of 160°F as it is for an equipment exit temperature of 275°F.

4) The total equivalent weight can be reduced somewhat (a rather significant amount for the more severe conditions) by use of a cascaded cycle (see figure 36); however, greater complexity is involved in this type of system.

5) Vapor cycle systems are not considered suitable for individualized cooling applications. The type of heat transport required, discussed in section IV, imposes a relatively minor equivalent weight.

2. Water Vapor Cycle Cooling Systems

The application of water as the refrigerant will result in numerous advantages over the more standard refrigerants; among the major advantages are

1) Systems using water as a refrigerant can operate over the entire Mach number-altitude envelope considered in this study. Systems would be designed for operating up to Mach number in the range of 2.8 to 3.0.

2) The total equivalent weight of water cycle systems is considerably less than for Freon systems for Mach numbers above 1.8. The system equivalent weight increases with velocity but at a much lower rate than does the Freon-11 system.

3) The total equivalent weight is somewhat less dependent on altitude and on equipment exit temperature than is the Freon system.

4) The water cycle system is relatively efficient as a heat pump for the entire range of conditions considered, i.e., the power input, while diverging somewhat from the Carnot value (see figure) follows a uniform trend and is at a very reasonable level throughout the temperature range that is of interest in this study. In fact, water is the only fluid noted that possesses such characteristics.

It should be noted that the use of water does entail several problems that require attention before working systems can be built. It is believed that the problems are of a nature that would yield to a moderate practical engineering effort. Among the factors requiring specific attention are the following:

1) Design of a system that is immune to freeze damage and is self-thawing.

2) Design of a compressor suitable for compression of water vapor.

3) Design of a temperature control system.

4) Design of a pressure control system.

Given the necessary engineering effort for development along the lines outlined above, the water vapor cycle cooling system could well become one of the better equipment cooling systems for long range, high speed aircraft.

SECTION VI

AIR CYCLE COOLING SYSTEMS

A. Basic Considerations

Because of the widespread current application of air cycle principles to cooland and air conditioning systems for aircraft and the probability that air cooling concepts will continue to be used in many further aircraft, careful attention has been directed toward a realistic analysis and evaluation of air cycle equipment cooling systems. The air cycle system is particularly attractive for aircraft equipped with turbojet engines since high pressure air bled from the engine compressor is a convenient source of air for the cycle. It is important to note in this regard that many new engines are being designed for operation with up to 7% compressor bleed, principally for cabin conditioning and pressurization. All air cycle systems utilizing compressor bleed air function essentially as follows: The bleed air is first pre-cooled in one (or more) heat exhanger(s) and then further cooled by passage through an expansion turbine, after which it is employed either directly or indirectly for cooling of equipment aboard the aircraft. Differences among the various types of air cycle systems result from different arrangements of heat exchangers and turbine, together with other system elements such as compressors (or blowers), etc. Specific systems which are being considered in this study are described and discussed in subsequent sections.

The altitude-Mach number range specified for the cooling system study is shown in figure 1. Curves of constant ram air temperature are shown in figure 3. The ram air temperature is one of the fundamental independent variables of the air cycle system analysis since it represents the temperature level of the most readily available thermal sink for removal of heat from the engine bleed air. Another fundamental system variable is the allowable equipment exit temperature (T_{E_e}) which for this study was specified in the range from 160° to 275°F. The major aim of the present study of air cycle cooling systems is to select and analyze in detail some of the typical air cycle cooling systems, for the specified range of conditions, in order to determine the relative merit of these selected systems and to compare them with other types of cooling systems. Evaluation criteria for the comparison of cooling systems are discussed in section III of this report.

B. Components of Air Cycle Cooling Systems

Air cycle cooling systems, in general, include the following basic components:

1) Heat exchangers are used to transfer heat from one fluid to another, normally air to air but for some applications air to liquid heat exchangers are needed.

2) Air turbines are used to reduce the temperature of the air by extracting mechanical energy.

3) A blower or a compressor is used to provide a load on the air turbine and to build up air pressure. The increase in available energy can be utilized by exhausting the air through a nozzle thus providing a thrust recovery.

4) Air ducts are needed to conduct the air to and from the cooling system. In addition to the above basic components, certain controls and pressure or flow regulating units are required to complete the system.

1. Heat Exchangers for Air Cycle Systems

As used in air cycle cooling systems, the heat exchanger is called upon to cool a quantity of hot air to a temperature approaching that of an available thermal sink. Since a large part of the cooling in an air cycle system occurs in the heat exchangers, it is usually desirable to employ exchangers with high effectiveness. However, this desirable characteristic can be obtained only at the expense of increased size and weight of the exchanger or by increased coolant flow. Thus, the selection of a heat exchanger for an air cycle system involves a compromise between the greater cooling achieved with high effectiveness and the increased sixe and weight of the exchanger or by increased coolant flow. Thus, the selection of a heat exchanger for an air cycle system involves a compromise between the greater cooling achieved with high effectiveness and the increased weight and momentum drag which this entails. This problem is discussed in the following pages.

The analysis of heat exchanger requirements can be most conveniently carried out in two distinct steps as follows: first, determination of the required heat transfer conductance for specified values of effectiveness e_b and flow ratio w_a/w_b and, second, analysis of core characteristics to determine the size, weight, and pressure losses for an exchanger having the required cooling effectiveness. An analysis of the characteristics of specific heat exchangers is presented in appendix V.

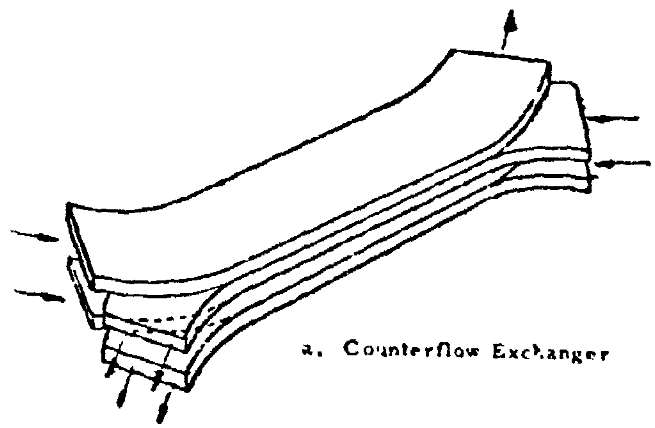
The literature contains a good deal of information relating to heat exchanger effectiveness to the conductance (UA), primary flow rate (w_1), coolant flow rate (w_2), and the geometrical arrangement of the exchanger. Theoretically, the counterflow arrangement (see figure 56a) affords the highest possible effectiveness for given values of UA, w_1 and w_2. Unfortunately, advantages of this arrangement cannot be realized in practice because of the difficulty of header design. The cross-flow arrangement shown in figure 56b eliminates the headering problems associated with the counterflow exchanger but has a lower effectiveness, other things being equal. The heat transfer charactersitics of the cross-flow heat exchanger were first determined by Nusselt (reference 16). Nusselt's work was based on the following assumptions:

1) The overall unit conductance (UA) between fluid streams is constant over the heat transfer surface.

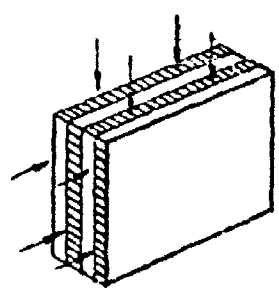
2) No mixing occurs in the individual fluid streams; at a local point, heat transfer to a fluid element of one stream occurs only by direct heat transfer from the other stream.

3) The respective fluid flow rates per unit passage area are constant over the heat transfer surface.

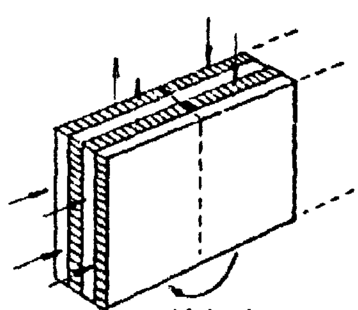
Nusselt's cross-flow formulae are given in reference 14 in an improved form more satisfactory for calculation, together with plotted results for use in engineering calculations. Cross-flow heat exchangers of higher effectiveness can be obtained by resorting to multi-pass arrangements as shown in figure 56c. Although no mathematically exact results based upon the assumptions of Nusselt's work have been reported for the multi-pass cross-flow exchanger, simplified methods of analysis can be used in most cases. One such method, based on the Nusselt cross-flow factor, is to apply the cross-flow reaults to successive passes, assuming complete mixing of the cooled and coolant fluid between passes. The fluid being cooled, which makes several passes through the exchanger, can be considered to mix thoroughly in the turning headers between passes; however, these is no physical basis for the assumption that the cooland mixes between passes. An analysis based on somewhat different assumptions was carried out during the present study. Here it was assumed that the fluid being cooled is maxed as it flows through the exchanger; no mixing being assumed for the coolant fluid. This assumption was based on the observation that the length of a single pass is usually several times its width.



a. Counterflow Exchanger



b. Single-Pass Crossflow Heat Exchanger



c. Multi-Pass Counter-Crossflow Heat Exchanger

FIGURE 56 TYPICAL HEAT EXCHANGER ARRANGEMENTS

Typical results of calculations using these various methods of analysis are shown in figure 57 for $e_b = 0.8$ and 0.9. The rpaid increase in required conductance (UA) with increasing effectiveness and decreasing flow ratio w_a/w_b is apparent regardless of the method of analysis employed. Comparison of the multi-pass results for $e_b = 0.8$ indicates that both methods of calculation agree quite closely for w_a/w_b greater than 1.5, but that considerable difference exists for small values of flow ratio with the analysis based on Nusselt's cross-flow factor leading to lower required conductance factors as would be expected. In interpreting these results, it is important to bear in mind the assumptions made in each case and the degree to which they are satisfied in an actual heat exchanger. It is known, for example, that a region of lowered conductance exists in the core for a short distance before the entering flow becomes fully turbulent. This is particularly true of heat exchangers having tubular cores (reference 17). Thus the conductance for the final pass is lowered (since the coolant enters at this point) and the effectiveness would also be reduced. Furthermore, the local overall conductance is dependent to a significant degree upon the mean temperature of the two fluid flows and consequently varies over the heat transfer surface (although the variation would probably not exceed 5-10%). The extent to which heat is transferred laterally through each fluid stream either by actual mixing of the turbulent streams (which would occur in flow over the tubes in a tube bundle core) or by metallic conduction (which would occur in a plate and fin core) would also exert a considerable influence upon the heat exchanger effectiveness. It is clear that all these effects tend to reduce the exchanger effectiveness. These considerations are especially significant in the case of unity flow ratio ($w_a/w_b = 1.0$) for which the effects of imperfect flow conditions are greatly magnified. For this reason, it is believed that the calculated results for mixed flow of the fluid being cooled are probably more satisfactory than the more idealized analysis based on Nusselt cross-flow factors. By using the theoretical results for double and triple pass exchangers with mixed conditions for the fluid being cooled, conductance factors were computed for $w_a/w_b = 1.0$. These results, which are used in the analysis of the regenerative system, are plotted in figure 58.

In general, the heat exchanger design conditions which may be regarded as independent variables are:

1) Heat exchanger effectiveness e_b

2) Respective flow rates of cooled and coolant fluids, w_b and w_a

3) The allowable pressure losses in the exchanger

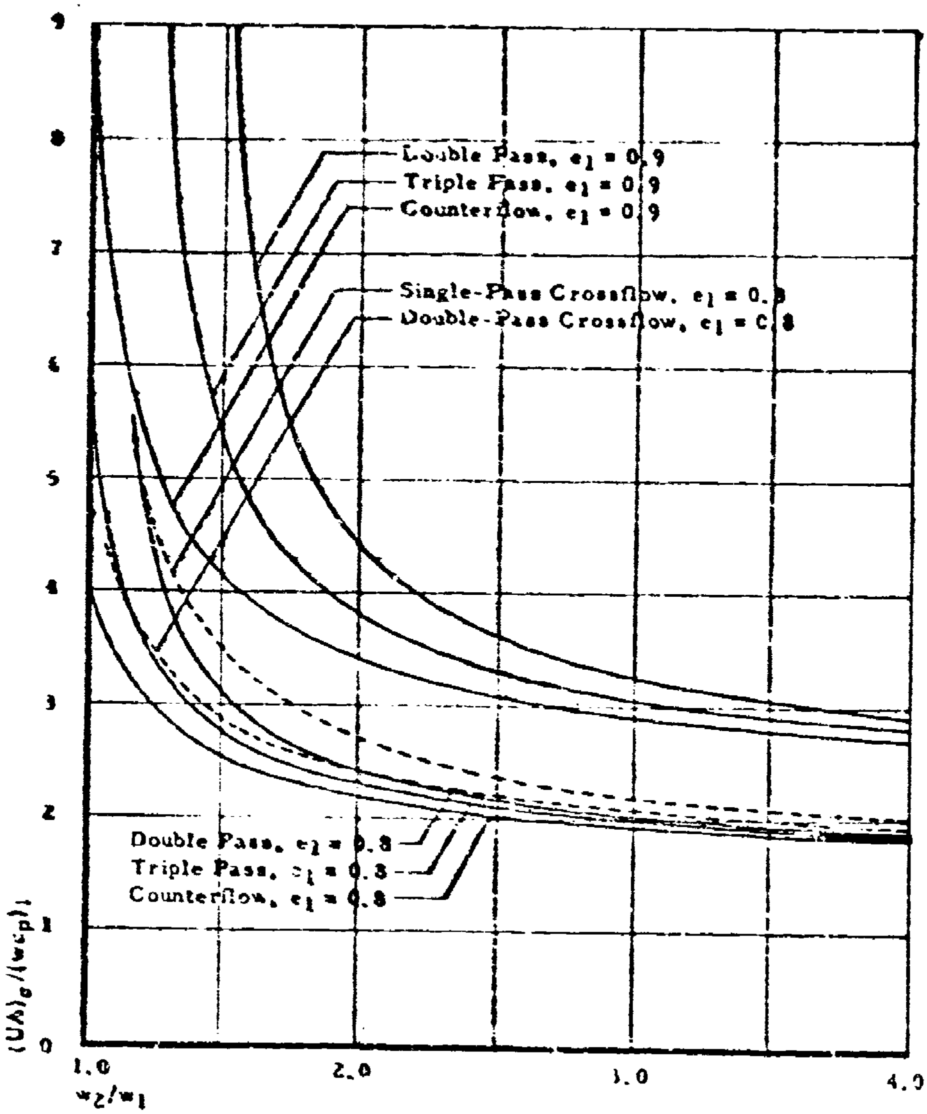


FIGURE 57 CROSSFLOW HEAT EXCHANGER PERFORMANCE

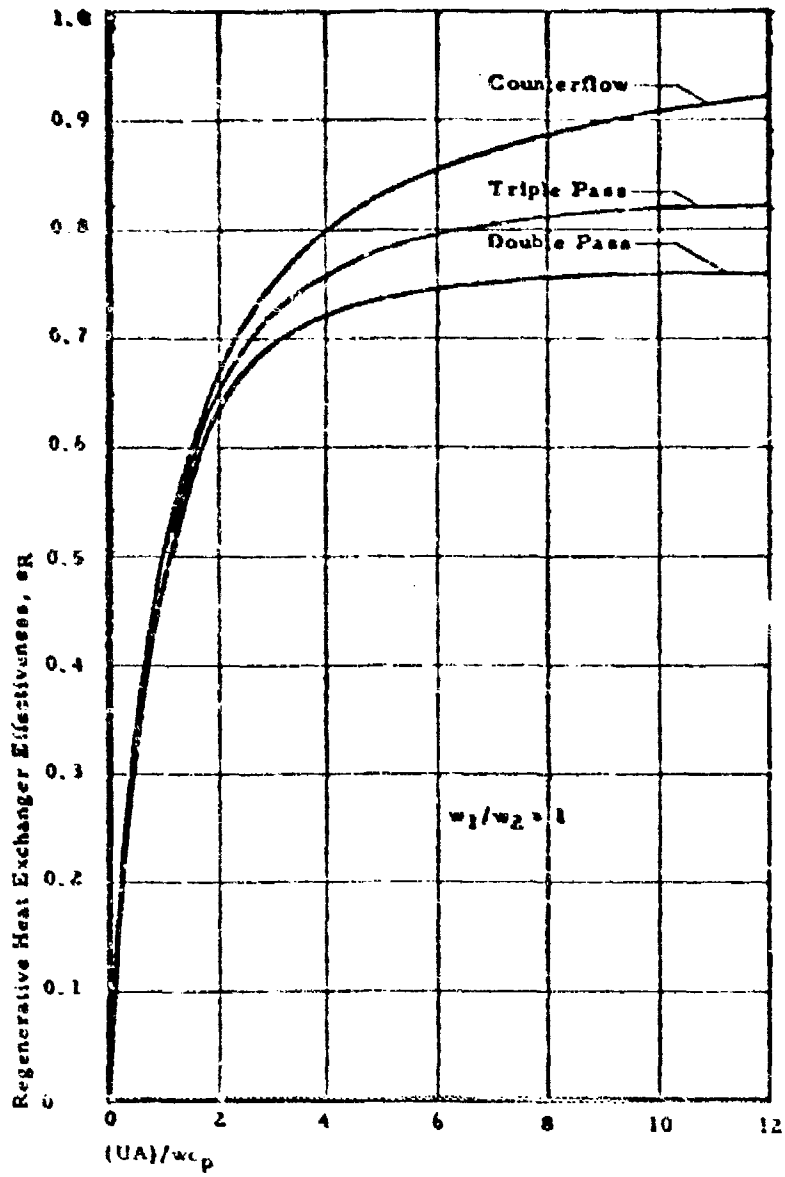


FIGURE 58 REGENERATIVE HEAT EXCHANGER PERFORMANCE

For a given type of core, the above conditions will determine the weight and size of the heat exchanger.

For applications involving moderate temperatures (under 600°F), aluminum heat exchanger cores are superior to stainless steel from the standpoint of weight. Because of the high bleed air temperatures, an aluminum pre-cooler is not applicable for the conditions of this study. However, an aluminum heat exchanger is applicable for use in the regenerative cycle such as described in section VI-E. The initial cooling of the bleed air being obtained by means of a stainless steel ram air heat exchanger or pre-cooler, subsequent regenerative cooling is at a temperature for which an aluminum core is suitable.

The characteristics and weights of air to air, liquid to air and boiling liquid type heat exchangers have been determined by the methods outlined in appendix I. These units are illustrated in figures 59, 33, and 60, respectively. The weights of ram air aluminum and stainless steel heat exchangers divided by the airflow rate is plotted versus a pressure drop parameter for effectiveness varying from 0.50 to 0.90 in figure 61. The weight factor versus the pressure drop parameter for an aluminum regenerative heat exchanger is plotted in figure 62. The weight of boiler heat exchanges divided by the airflow rate is plotted versus a pressure drop parameter in figure 63.

2. Expansion Turbine

In order to carry out a numerical analysis and evaluation of the air cycle cooling system, it is necessary to know the bleed air temperature and pressure as well as the aircraft performance effect due to compressor air extraction. The regenerative cooling system is inherently well adapted to equipment cooling for supersonic cruising conditions at $M = 2.0$ or above; consequently, engine characteristics compatible with efficient cruising performance at supersonic speeds were assumed in the analysis. Lacking detailed information on supersonic turbojet engines, the following assumptions were made regarding engine performance for extended supersonic cruising conditions.

1) Engine inlet pressure recovery based upon 85% diffuser efficiency for all conditions.

2) Engine compression ratio is 8.0 provided the compressor discharge temperature does not exceed 1250°R. In this regime, the turbine inlet temperature is assumed to be approximately double the compressor discharge temperature.

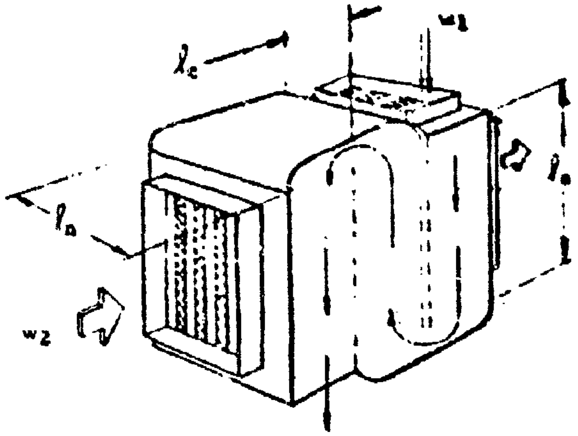


FIGURE 59 THREE-PASS AIR-TO-AIR HEAT EXCHANGER

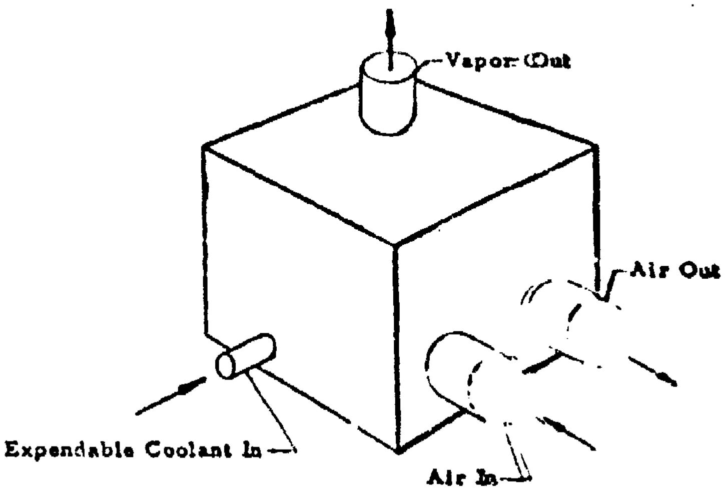


FIGURE 60 BOILER HEAT EXCHANGER

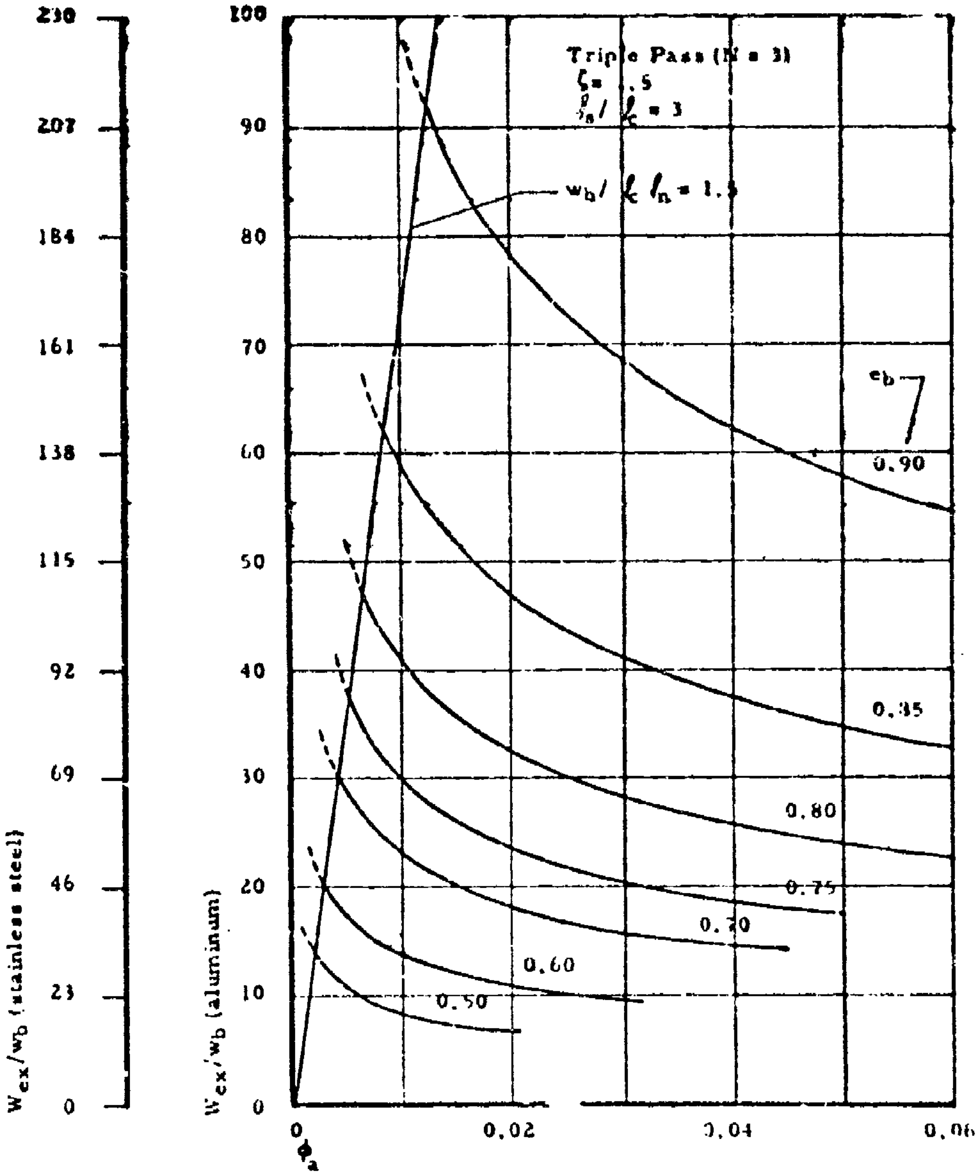


FIGURE 61 HEAT EXCHANGER WEIGHT

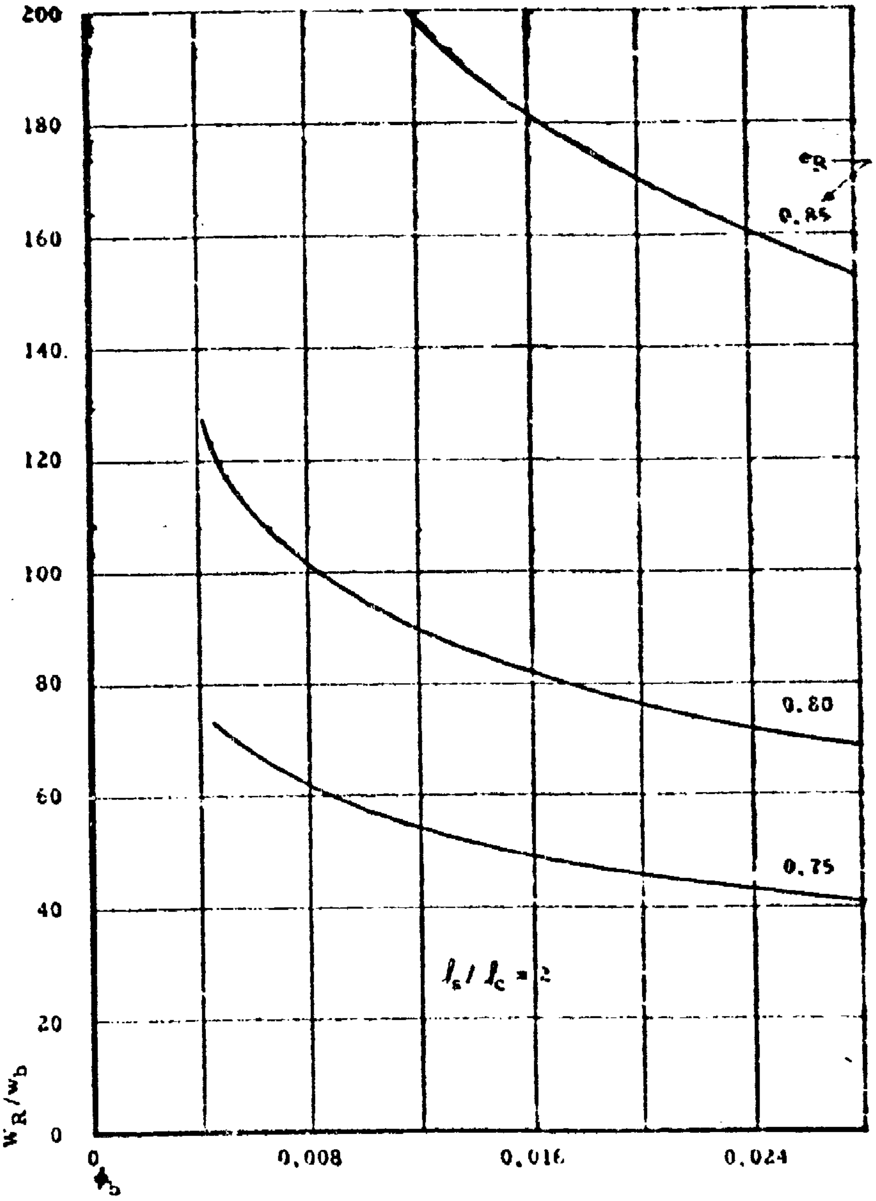


FIGURE 62 REGENERATIVE HEAT EXCHANGER WEIGHT (ALUMINUM EXTENDED-SURFACE CORE)

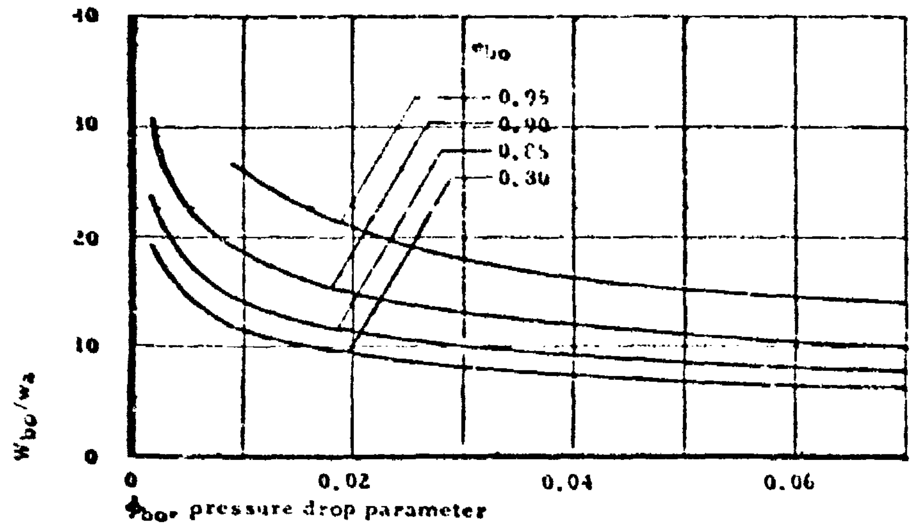


FIGURE 63 WEIGHT OF A BOILER HEAT EXCHANGER

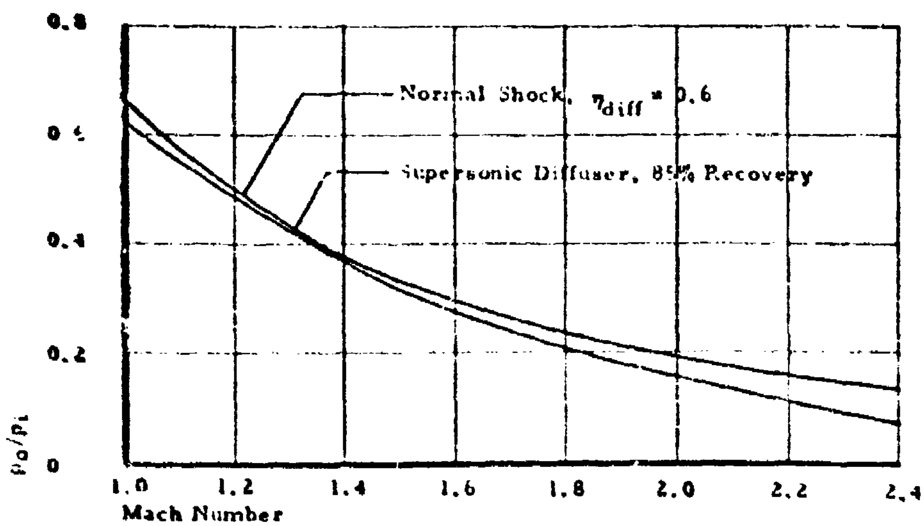


FIGURE 64 DUCT INLET PRESSURE RATIO VERSUS MACH NUMBER

3) For conditions not covered by item 2, the compressor discharge temperature is 1250°R, with an engine turbine inlet temperature of 2500°R.

4) For all conditions, the compressor efficiency is assumed to be 88%; the turbine efficiency is assumed to be 85%.

Of course, no given engine would operate in accordance with these conditions over a wide range of Mach number; however, an airplane designed for extended cruising flight at high speed (e.g., M = 2.0) would in all probability be fitted with an engine which would approach the above operating conditions at the design cruising speed. The assumed values of turbine inlet temperature and efficiency are included to enable estimation of the aircraft performance penalty due to compressor air extraction which is discussed below.

With the air cycle cooling systems being considered in this section of this report, the equipment is cooled by circulation of the cooling air through the equipment compartment. The engine bleed air used for cooling is first cooled by a ram air heat exchanger, discussed in the preceding section and in appendix I. The final cooling is secured by means of an air turbine which cools the air by extraction of work during expansion.

The amount of cooling that can be obtained during expansion depends on the amount of work done by the turbine per pound of air. That work in turn is dependent on the pressure or expansion ratio and on the turbine efficiency.

The turbine exit temperature is given by the equation

$$T_{te} = T_{t-i}\left\{1 - \eta_t \left[1 - \left(\frac{p_{ti}}{p_{te}}\right)^{0.286}\right]\right\} \tag{90}$$

The turbine inlet temperature is dependent on the engine bleed temperature, the pre-cooler effectiveness, and the ram air temperature.

$$T_{ti} = e_b\, T_T + (1 - e_b)\, T_b \tag{91}$$

The heat exchanger effectiveness can be considered as a design variable. The ram air temperatures (T_T) for the altitude-Mach number range of interest in this study are plotted in figure 65. The turbine bleed temperature depends on the ambient conditions and on the engine compressor characteristics. The variation of the normalized ambient temperature (θ_a) and the normalized ambient pressure ($_a$) assumed for this study are also shown in figure 65.

It is assumed in this study that 1250°R is the maximum compressor bleed temperature. The normalized compressor bleed temperature (θ_b) assumed for this study is plotted versus Mach number for various altitudes in figure 66.

The pressure ratio available for expansion across the turbine is determined by the bleed pressure, the pressure losses through the ducts and the heat exchanger, and the discharge pressure.

The assumed normalized bleed pressures ($_b$) are plotted versus Mach number for various altitudes in figure 67. If it is assumed that the discharge pressure is greater than ambient, e.g., to the aircraft cabin, the pressure ratio available for turbine expansion decreases rapidly with altitude. The ideal pressure ratios (for various engine compression ratios) neglecting duct and heat exchanger pressure drops, are plotted versus altitude in figure 68 assuming a discharge pressure of 7.5 psi above ambient. The ideal pressure ratios assuming a discharge pressure 2.7 psi above ambient are plotted in figure 69. These pressure ratios, together with the ram air temperature, impose definite altitude and Mach number limitations on simple air cycle cooling systems.

The expansion turbine is assumed to operate at pressure ratios up to 6:1. Operation at greater pressure ratios is usually accompanied by a reduced efficiency.

To eliminate the possibility of icing problems, the turbine exit temperature is not permitted to drop below 32°F.

If the heat exchanger effectiveness and the pressure drop for the heat exchanger and for the ducts are defined, the turbine exit temperature can be readily determined by equations (90) and (91), taking account of the assumed pressure and temperature limitations outlined above.

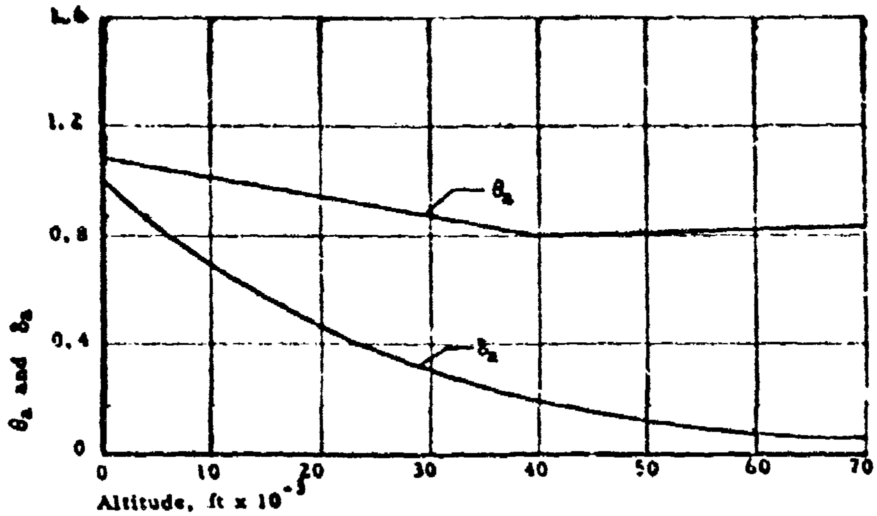


FIGURE 65 VARIATION OF AMBIENT PRESSURE AND TEMPERATURE WITH ALTITUDE

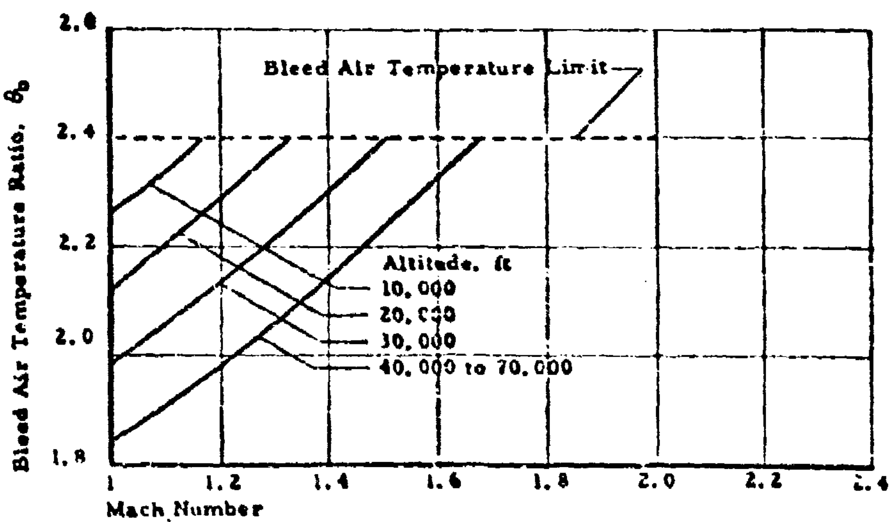


FIGURE 66 BLEED AIR TEMPERATURE RATIO VERSUS MACH NUMBER

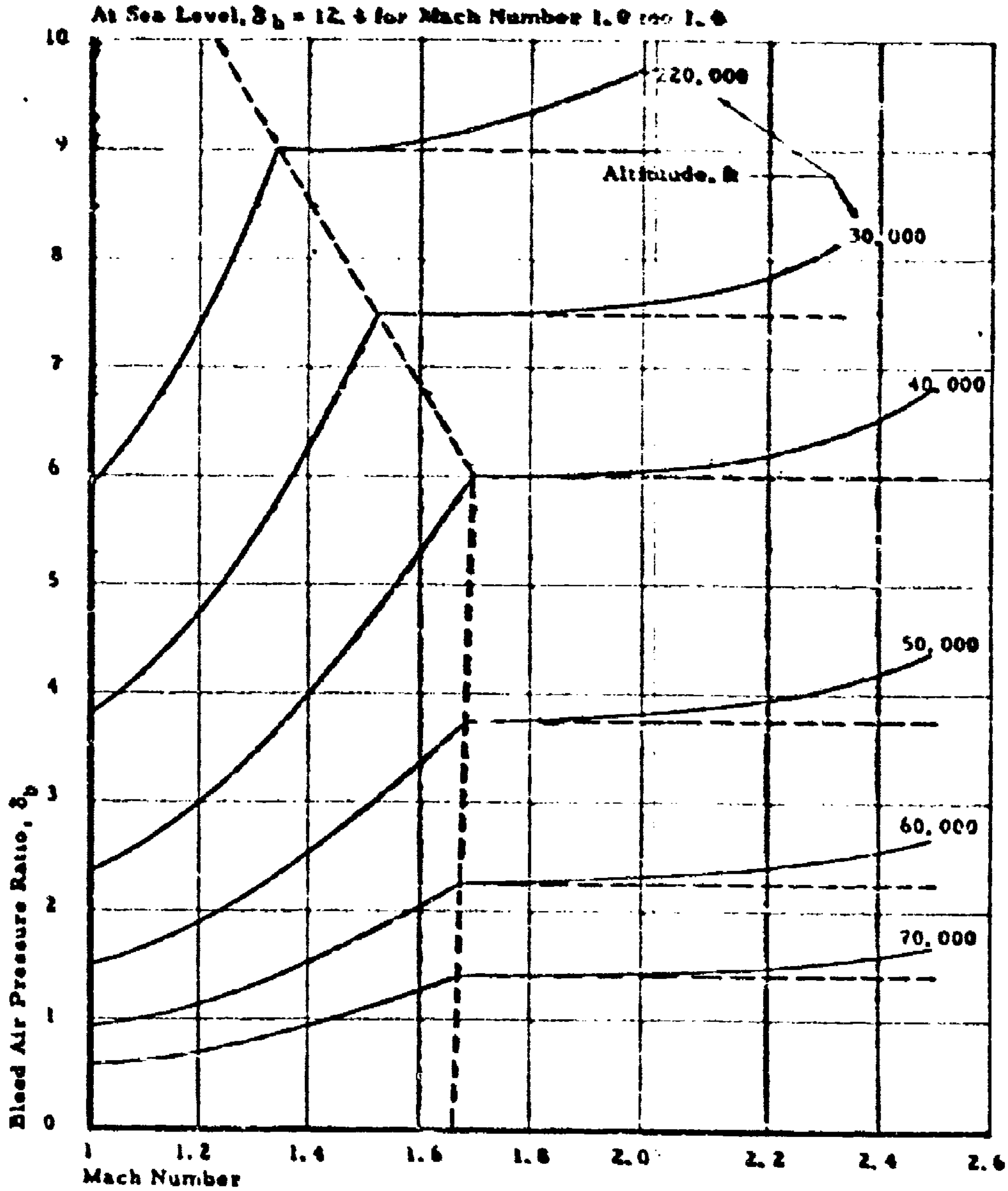


FIGURE 67 BLEED AIR PRESSURE RATIO VERSUS MACH NUMBER AND ALTITUDE

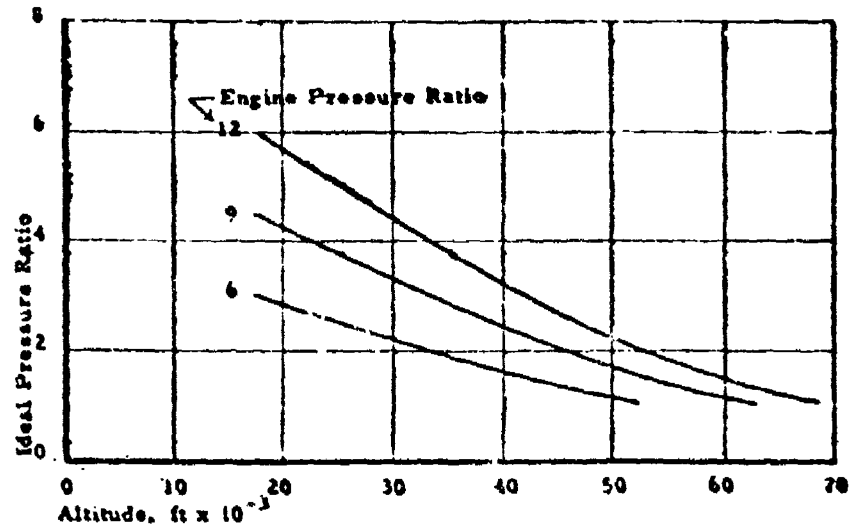


FIGURE 68 IDEAL PRESSURE RATIO VERSUS FLIGHT ALTITUDE WITH A DISCHARGE PRESSURE OF 7.5 psi ABOVE AMBIENT

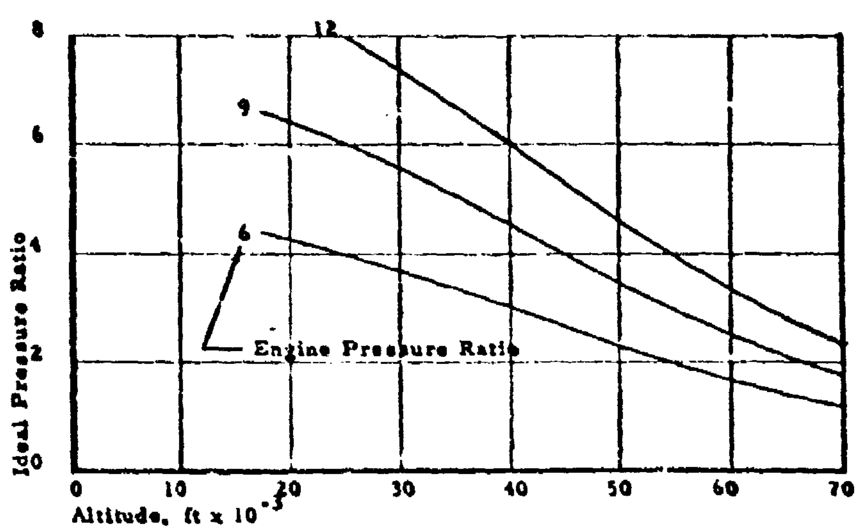


FIGURE 69 IDEAL PRESSURE RATIO VERSUS FLIGHT ALTITUDE WITH A DISCHARGE PRESSURE OF 2.7 psi ABOVE AMBIENT

In addition to determination of the temperatures that can be obtained with an air turbine, the analysis of turbines for air cycle cooling systems requires reasonably accurate determination of turbine weights. A detailed analysis of the characteristics of turbines of the type used for air cycle cooling systems indicates that the weight of such units (including the blower discussed below) can be very closely approximated by a factor times the airflow rate. The required airflow rate is dependent on the cooling load and the temperature difference.

$$w_b = \frac{3.95\ kw}{T_{Ee} - T_{Ei}} \tag{92}$$

A good approximation to the turbine weight is given by the empirical equation

$$W_t = 24\ w_b \tag{93}$$

It can be reasonably assumed that the turbine and blower weight will not be less than eight pounds; consequently, eight pounds is taken as a minimum weight and is used for all cases where w_b 0.333 lbs/sec.

3. Blowers and Compressors

The taking of ram air on board an aircraft for heat exchanger cooling involves a momentum drag due to the reduction in the air velocity. Part of that drag can be recovered by discharging the air through a thrust recovery nozzle. A blower is usually utilized to increase the pressure of the ram air before discharge thus increasing the thrust recovery factor. The blower also serves as a load for the expansion turbine.

The drag due to taking air on board the aircraft is given by

$$D = w_a\ (34.7\sqrt{\theta_a}\ M) \tag{94}$$

The thrust recovery is dependent on the nozzle pressure ratio, the temperature, and the nozzle efficiency; for supercritical nozzle flow (p_a/p_{ne} 0.528).

$$T = w_a\ 54.4\ \eta_{ne}\ \sqrt{\theta_{ne}} \left(1 - 0.788\ \frac{p_a}{p_{ne}}\right) \tag{95}$$

For subcritical nozzle flow

$$T = w_a\ 78.8\ \eta_{ne}\ \sqrt{\theta_{ne}}\ \left[1 - \left(\frac{p_a}{p_{ne}}\right)^{0.286}\right]^{1/2} \tag{96}$$

The net momentum drag is then

$$D_{mom} = w_a\quad 34.7\ \sqrt{\theta_a}\ M - F_{th}\ \sqrt{\theta_{ne}} \tag{97}$$

Where for supercritical flow

$$F_{th} = 54.4\ \eta_{ne}\ \left(1 - 0.788\ \frac{p_a}{p_{ne}}\right)$$

when $p_a/p_{ne} \leq 0.528$.

For subcritical flow

$$F_{th} = 78.8\ \eta_{ne}\left[1 - \left(\frac{p_a}{p_{ne}}\right)^{0.286}\right]^{1/2}$$

when $p_a/p_{ne} \geq 0.528$.

In most air cycle systems a blower or compressor is used to provide a load on the air turbine and to increase the pressure and temperature of the air before it is exhausted. From the above equations, it is apparent that the thrust recovery may be improved by increasing the pressure, by increasing the temperature, by keeping the pressure losses at a minimum, and by using an efficient nozzle.

In most air cycle cooling systems, a blower or compressor is used to provide a load for the air turbine and to increase the pressure and temperature upstream from the exhaust nozzle thus increasing the thrust recovery.

The pressure ratio for a blower of conventional design can be assume to be about 1.1. The rise in temperature is determined by the decrease obtained with the turbine. The pressure ratio for a compressor can be expressed in terms of the adiabatic compressor efficiency and inlet and exit temperatures by the equation

$$r_c = \left(1 + \eta_c\ \frac{\theta_{ci}}{\Delta\theta_c}\right)^{3.5} \tag{98}$$

The temperature increase produced by the compressor is related to the cooling produced by the expansion turbine, neglecting mechanical losses.

$$\Delta\theta_c = \frac{w_t}{w_c} \Delta\theta_t \tag{99}$$

The nozzle exit temperature can be defined by

$$\theta_{ne} = \theta_T + \frac{\epsilon_b}{\zeta} (\theta_b - \theta_T) + \Delta\theta_c \tag{100}$$

The net drag associated with the ram air can be determined by means of the applicable equations as outlined above.

4. Air Ducts

As pointed out in the discussion on expansion turbines, the temperature drop that can be obtained by expansion through an air turbine is decreased by pressure losses in the ducts and heat exchangers. The lower such pressure losses are, the higher the pressure ratio available for the turbine cooling. However, a decrease in pressure drops entails an increase in size and weights of ducts and heat exchangers (in the case of heat exchangers, very low pressure drops are also associated with low heat transfer coefficients which are of course undesirable). Heat exchanger design is discussed in section
This section will be limited to a consideration of the ducts; however, the effects on all components and overall system performance must be borne in mind at all times. Because of the interaction of the effects, a compromise must be reached. Mathematical optimization, however, is not deemed practical. In lieu of a very complex optimization analysis, which could not in practice be rigorous, an engineering approach has been used in this study. The size and weights of the ducts are defined in terms of an allowable pressure drop, which is selected on the basis of system operating requirements.

$$\frac{\Delta P}{P_i} = 0.5\, F_{dt}\, f\, w^{1.8} \frac{\theta}{\delta^2}\, d^{-4.8} \tag{101}$$

The required duct diameter is then

$$d = \left(0.5\, F_{dt}\, f\, w^{1.8} \frac{\theta}{\delta^2} \frac{P_i}{\Delta P}\right)^{0.208} \tag{102}$$

In equations (101) and (102) the factor F_{dt} is an adjustment factor to account approximately for increased losses in the inlet region of the duct and other losses, e.g., diffuser losses at the entrance to a heat exchanger. The factor F_{dt} varies from one, for fully developed flow, to approximately 2.5 for relatively high losses.

The weight of the ducts can be determined from the length and weight per foot for the required size with an allowance for the necessary fittings. The weight of representative ducts can be expressed by the equation

$$W_{dt} = 0.13\, d\, \ell \qquad (103)$$

The inlet pressure to a ram air duct from a supersonic diffuser with an 85% recovery can be determined by means of figure 64.

C. Effects of Engine Compressor Bleed

The air cycle cooling systems considered in this study utilize bleed air from the engine compressor as the cooling fluid. The amount of air that is required will depend on the equipment inlet and exit temperatures. The exit temperature is considered an independent design factor. The equipment inlet temperature is then determined by the heat exchanger effectiveness, turbine pressure ratio, and turbine efficiency. These factors vary with system design and with flight conditions. The amount of ram air required and its effect is discussed along with the consideration of air ducts.

A comparison of systems that impose varying air bleed and, further, a comparison of such systems with systems that do not utilize any bleed air, requires the determination of the effect of that bleed on the engine thrust. An equivalent weight can then be determined by the application of a translation factor as discussed in section III.

The effect of bleeding air from an engine compressor can be considered in two somewhat different ways:

1) The engine compressor is basically designed for adequate air bleed so that any air that is bled would not be considered as potentially available for burning of fuel and the production of engine thrust; or

2) the air bled would otherwise be mixed with fuel which could then burn and produce thrust.

The first viewpoint has been chosen and used in this analysis. With this assumption, the extraction of bleed air results in a decrease in engine thrust from (1) a momentum drag associated with taking the air on board the aircraft and (2) the energy required to compress the bleed air in the engine compressor. The compression of the air also results in an increase in the specific fuel consumption in as much as the energy expended in compressing the bleed air is not available for thrust.

The alternate assumption, that the bleed air would have been used in burning fuel and generating thrust, would result in a slightly greater significance being assigned to air bleed because of the reduction in net thrust.

Assuming that the engine compressor is basically designed for adequate bleed, the effects have been analyzed by employing the procedure of reference 18.

The gross thrust of the engine working fluid is approximately

$$T_G \doteq (T_G)_{w_b = 0}\left[1 - \frac{w_b}{W_T}\left(\frac{\Delta h_C}{2\Delta h_n}\right)_{w_b = 0}\right] \tag{104}$$

where $\left(\frac{\Delta h_C}{\Delta h_n}\right)_{w_b = 0}$

is the ratio of the enthalpy increase through the compressor to the enthalpy decrease through the exhaust nozzle evaluated at zero air bleed. The gross thrust change is approximately

$$\frac{\Delta T_G}{(T_G)_{w_b = 0}} \doteq \frac{-w_b}{W_T}\left(\frac{\Delta h_C}{2\Delta h_n}\right)_{w_b = 0} \tag{105}$$

The change in net thrust is then

$$\frac{\Delta T_N}{(T_N)_{w_b = 0}} \doteq \frac{-w_b}{W_T}\left(\frac{T_G}{T_N}\,\frac{\Delta h_C}{\Delta h_n}\right)_{w_b = 0} \tag{106}$$

The specific fuel consumption is

$$SFC = (SFC)_{w_b = 0}\;\frac{1}{1 - \frac{w_b}{W_T}\left(\frac{T_G}{T_N}\,\frac{\Delta h_C}{2\Delta h_n}\right)_{w_b = 0}} \tag{107}$$

$$\frac{\Delta SFC}{(SFC)_{w_b = 0}} = \frac{w_b}{W_T}\left(\frac{T_G}{T_N}\,\frac{\Delta h_C}{2\Delta h_n}\right)_{w_b = 0} \tag{108}$$

The net penalty for the working fluid in terms of thrust reduction can now be determined by adding the above fractional effects and multiplying through by the net thrust (T_N). (The actual thrust change due to compressing the bleed air is negative.)

$$\Delta T_N = -w_b \left(\frac{T_N}{W_T}\,\frac{T_G}{T_N}\,\frac{\Delta h_C}{2\Delta h_n}\right)_{w_b = 0} \tag{109}$$

The term T_N/W_T is the specific thrust. The momentum drag associated with taking the bleed air on board the aircraft is

$$D = 34.7 w_b M \sqrt{\theta_a} \tag{110}$$

The total change in thrust due to engine bleed is then

$$\Delta T_{bT} = w_b \left[34.7 M \sqrt{\theta_a} + \left(\frac{T_N}{W_T}\,\frac{T_G}{T_N}\,\frac{\Delta h_C}{2\Delta h_n}\right)_{w_b = 0} \right] \tag{111}$$

$$= w_b\, F_{th\text{-}b} \tag{111a}$$

This negative thrust effect can be treated as a drag and an equivalent wieght determined by the methods outlined in section III. The first term in the above expression for $F_{th\text{-}b}$ corresponds to the momentum drag caused by taking the bleed air aboard the aircraft, while the second accounts for the direct effects of thrust loss and increased specific fuel consumption resulting from compression of the bleed air in the engine. In equation (111), the factor T_N/W_T is the engine specific thrust, T_G/T_N is the ratio of gross thrust to net thrust, and h_C/h_n is the ratio of the enthalpy increase across the compressor to the enthalpy decrease in the exhaust nozzle, each of these factors being evaluated for zero compressor bleed. Numerical evaluation of the compressor bleed penalty factor $F_{th\text{-}b}$ was carried out by using charts supplied in reference 18 together with the assumed turbojet operating conditions given previously. The results disclosed that $F_{th\text{-}b}$ was substantially constant over the range of flight conditions involved in this study, thus, it was assumed that

$$F_{th\text{-}b} = 120.$$

independent of Mach number and altitude. (The maximum deviation from the above value in the computed results was under 5%.)

D. Simple Air Cycle Cooling Systems

1. Method of Analysis and Assumptions

The simple air cycle cooling system is illustrated schematically in figure 70. In this system, engine bleed air (the cooling fluid) is cooled initially by passage through a bleed air to ram air heat exchanger. The bleed air is then further cooled by expansion through an air turbine which extracts energy in the form of mechanical work. The engine bleed air is then used for direct cooling of the equipment. The turbine shaft power is used to drive a blower or compressor which increases the pressure of the ram air before it is expelled overboard. The ram air is discharged through a nozzle designed to produce thrust and thus recover drag incurred when taking the ram air on board. The simple air cycle has been analyzed for the following conditions:

$T_{Ee} = 275°F$	equipment exit temperature
$M = 1 - 2$	flight velocity
$H = 0\text{-}70{,}000$ feet	altitude
$kw = 10$ kilowatts	cooling load

The assumed ambient and engine bleed pressures and temperatures are indicated in figures 65, 66, and 67. Other assumptions are indicated below. The effects of the various factors were investigated. The assumptions as listed below are considered reasonable and result in approximately the optimum values.

The amount of air (w_b) that must be bled from the engine compressor is dependent on the cooling load and the difference between the equipment inlet temperature (T_{Ei}) and the equipment exit temperature (T_{Ee}). The bleed rate is expressed by equation (92). The equipment inlet temperature is equal to the turbine exit temperature (T_{te}) which in turn is determined by the turbine pressure ratio (r_t) and its efficiency (η_t). The turbine exit temperature is given by equation (90).

Combining equations (90) and (92) and dividing through by T_{Ee}

$$w_b = \frac{3.95 kw / T_{Ee}}{1 - \frac{T_{t1}}{T_{Ee}} \left[1 - \eta_t \left(1 - r_t^{-0.286}\right)\right]} \tag{112}$$

The turbine inlet temperature is dependent on the compressor bleed temperature, the total ram temperature, and the effectiveness of the heat exchanger. That temperature is given by equation (91).

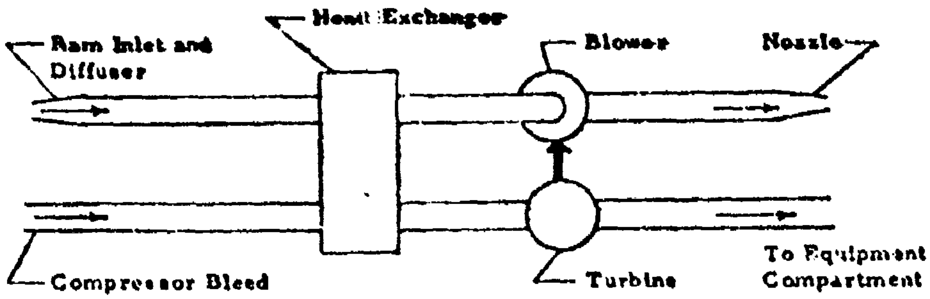


FIGURE 70 SIMPLE AIR CYCLE COOLING SYSTEM

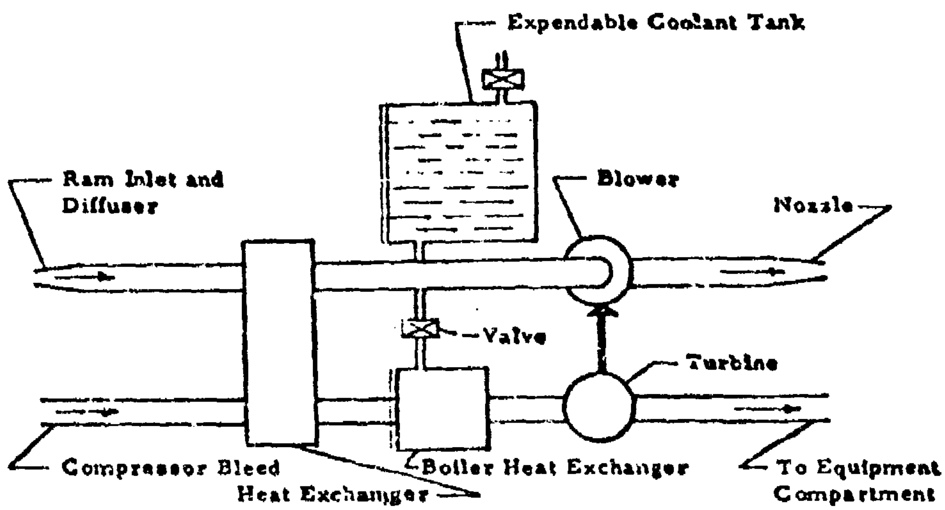


FIGURE 71 COMBINATION AIR CYCLE AND EXPENDABLE COOLANT SYSTEM

The heat exchanger effectiveness (e_b) is assumed to be 0.85 except for those cases at high altitudes where the turbine pressure ratio (r_t) is small. The effectiveness is reduced so that $w_b/\zeta_c \zeta_n$ does not drop below 1.5. This assumption is to assure turbulent air flow in the heat exchanger. The flow ratio (ζ) is taken as 1.5 for all cases. The pressure drop parameter (ϕ_b) is taken as

$$\phi_b = \frac{0.0024}{\theta_a} \qquad \text{for } \delta_i \quad 0.156 \tag{113}$$

$$\phi_b = \frac{\delta_i^{\,2}}{10\,\theta_a} \qquad \text{for } \delta_i \quad 0.156 \tag{113a}$$

$$\text{where } \theta_a = \theta_T + \frac{e_b}{2\zeta}(\theta_b - \theta_T) \tag{114}$$

The pressure drop of the cooling air is

$$\frac{\Delta p}{p_i} = \frac{0.0024}{\delta_i^{\,4}} \qquad \text{for } \delta_i \leq 0.156 \tag{115}$$

$$\frac{\Delta p}{p_i} = 0.10 \qquad \text{for } \delta_i > 0.156 \tag{115a}$$

To preclude icing problems, the turbine outlet temperature is not allowed to drop below 32°F. This is done by maintaining the bleed air rate at a minimum value such that, with the given equipment exit temperature, the turbine exit will not be below 32°F.

$$T_{te} = T_{Ei} = T_{Eo} - \frac{3.95\,kw}{w_a} > 32\text{°F} \tag{116}$$

The ducts from engine to heat exchanger and from turbine to equipment are assumed to be 10 feet long and are analyzed in section VI-B. The duct pressure drop parameter is assumed to be $\phi_{dt} = 0.02$ for each duct. The required diameter is given by equation (102). The assumed value of the factor F_{dt} is 2 for the duct from engine to heat exchanger to allow for losses at the heat exchanger inlet. $F_{dt} = 1$ for the duct from the turbine to the equipment.

The turbine pressure ratio is defined in terms of the "ideal" pressure ratio (r_o) with the discharge pressure maintained at 2 psi above ambient. Then

$$r_o = \frac{\theta_b}{\delta_a\left(1 + \frac{\Delta P}{P_a}\right)} \qquad (117)$$

For those cases where $r_o > 6$, the actual turbine pressure ratio is assumed to be maintained at 6 in this analysis. When $r_o < 6$, the turbine pressure ratio is assumed to be

$$r = 0.96 r_o \left(1 - \frac{0.025}{\delta_b^2} \frac{\bar{\theta}_b}{\bar{\theta}_a}\right) \qquad (118)$$

where $\bar{\theta}_a$ is given by equation (114) and

$$\bar{\theta}_b = \theta_b - \frac{e_b}{2} (\theta_h - \theta_T) \qquad (119)$$

The turbine and blower weight is given by equation (93) for $w_b > 0.333$. For $w_b < 0.333$, the turbine and blower weight is assumed to be 8 pounds.

The ram air duct is assumed to be 12 feet long with a pressure drop of 0.05. The factor F_{dt} for this duct is assumed to be 2.5 to allow for diffuser exit and heat exchanger inlet losses.

The effects of compressor bleed were analyzed as discussed in section In final calculations, the bleed factor is $F_b = 120$.

Ram air momentum drag is determined according to equation (97). The blower pressure ratio is assumed to be 1.1. The nozzle exit temperature is given by equation (100).

2. Results of the Analysis of Simple Air Cycle Cooling Systems

The total equivalent weights for the simple air cycle cooling systems are plotted versus Mach number for various altitudes in figure 72 for a drag translation factor $f = 2$ and in figure 73 for a drag translation factor $f = 3$. The total equivalent weight versus altitude is shown in figures 74 and 75. The operating limits for a simple air cycle cooling system of various total equivalent weights are shown on the altitude and Mach number envelope in figures 76 and 77.

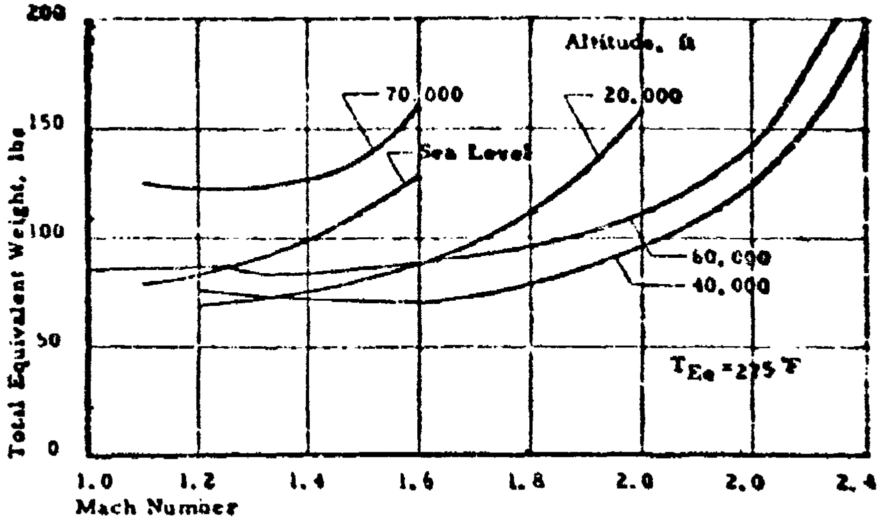


FIGURE 72 TOTAL EQUIVALENT WEIGHT VERSUS MACH NUMBER OF A SIMPLE AIR CYCLE COOLING SYSTEM (f 2)

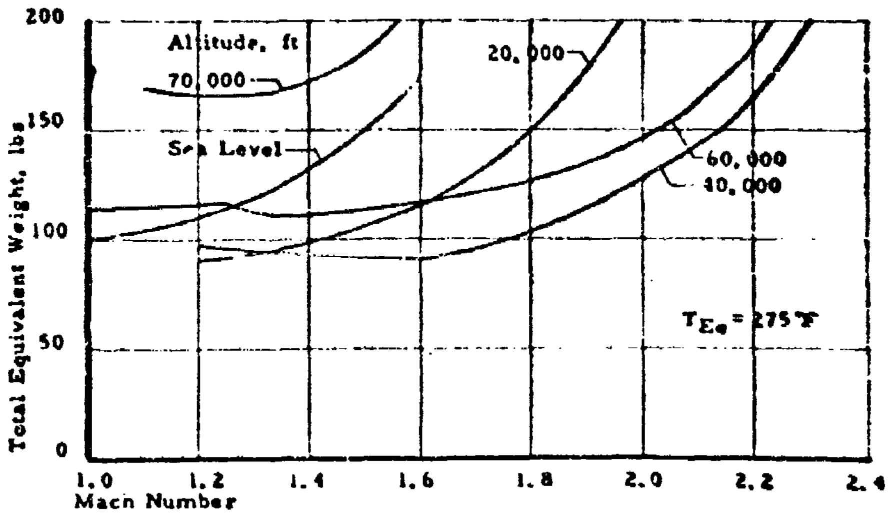


FIGURE 73 TOTAL EQUIVALENT WEIGHT VERSUS MACH NUMBER OF A SIMPLE AIR CYCLE COOLING SYSTEM (f 3)

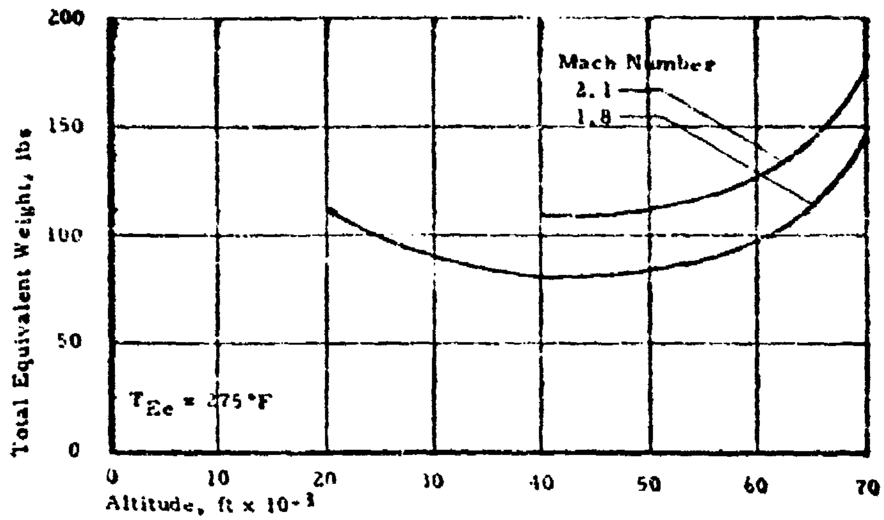


FIGURE 74 TOTAL EQUIVALENT WEIGHT VERSUS ALTITUDE OF A SIMPLE AIR CYCLE COOLING SYSTEM (f = 2)

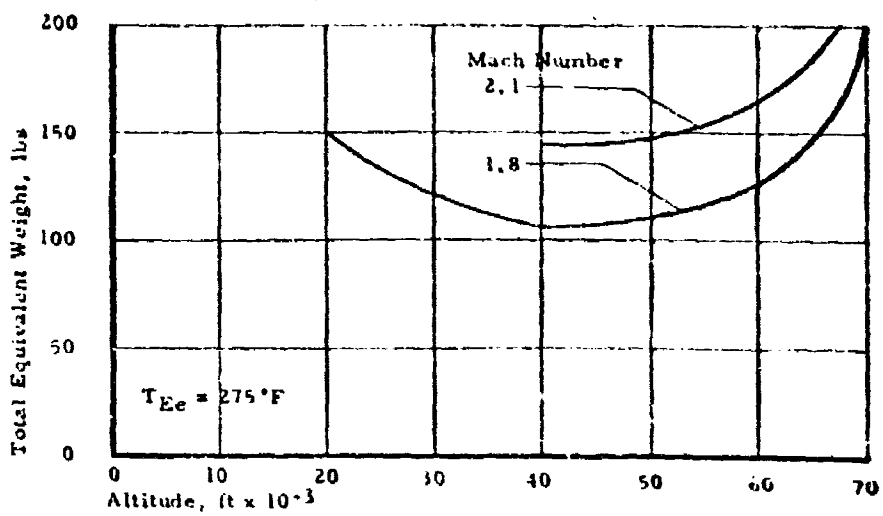


FIGURE 75 TOTAL EQUIVALENT WEIGHT VERSUS ALTITUDE OF A SIMPLE AIR CYCLE COOLING SYSTEM (f = 3)

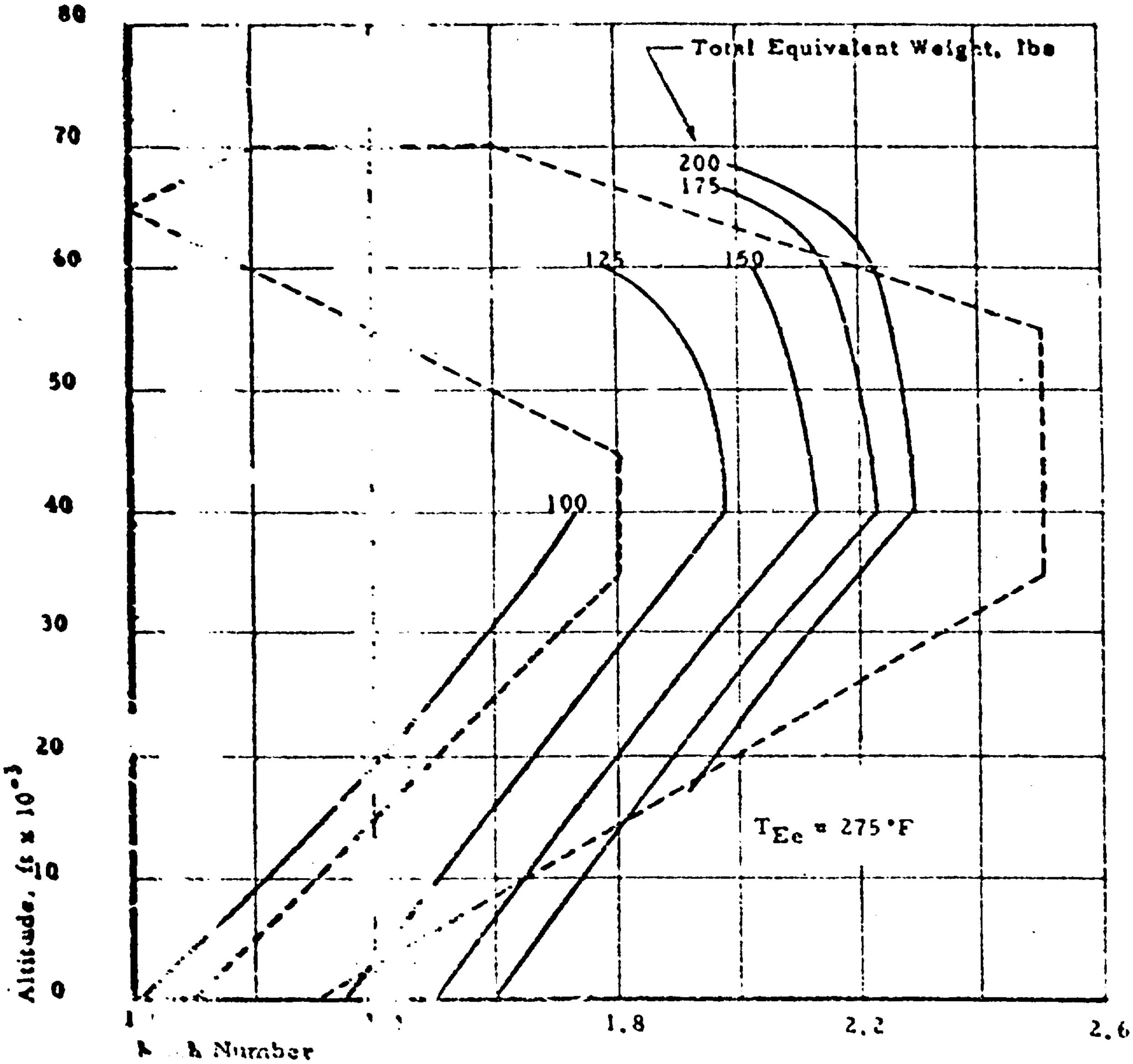


FIGURE 75 COOLING SYSTEM OPERATING LIMITS FOR A GIVEN TOTAL EQUIVALENT WEIGHT FOR A SIMPLE AIR CYCLE COOLING SYSTEM (f = 3)

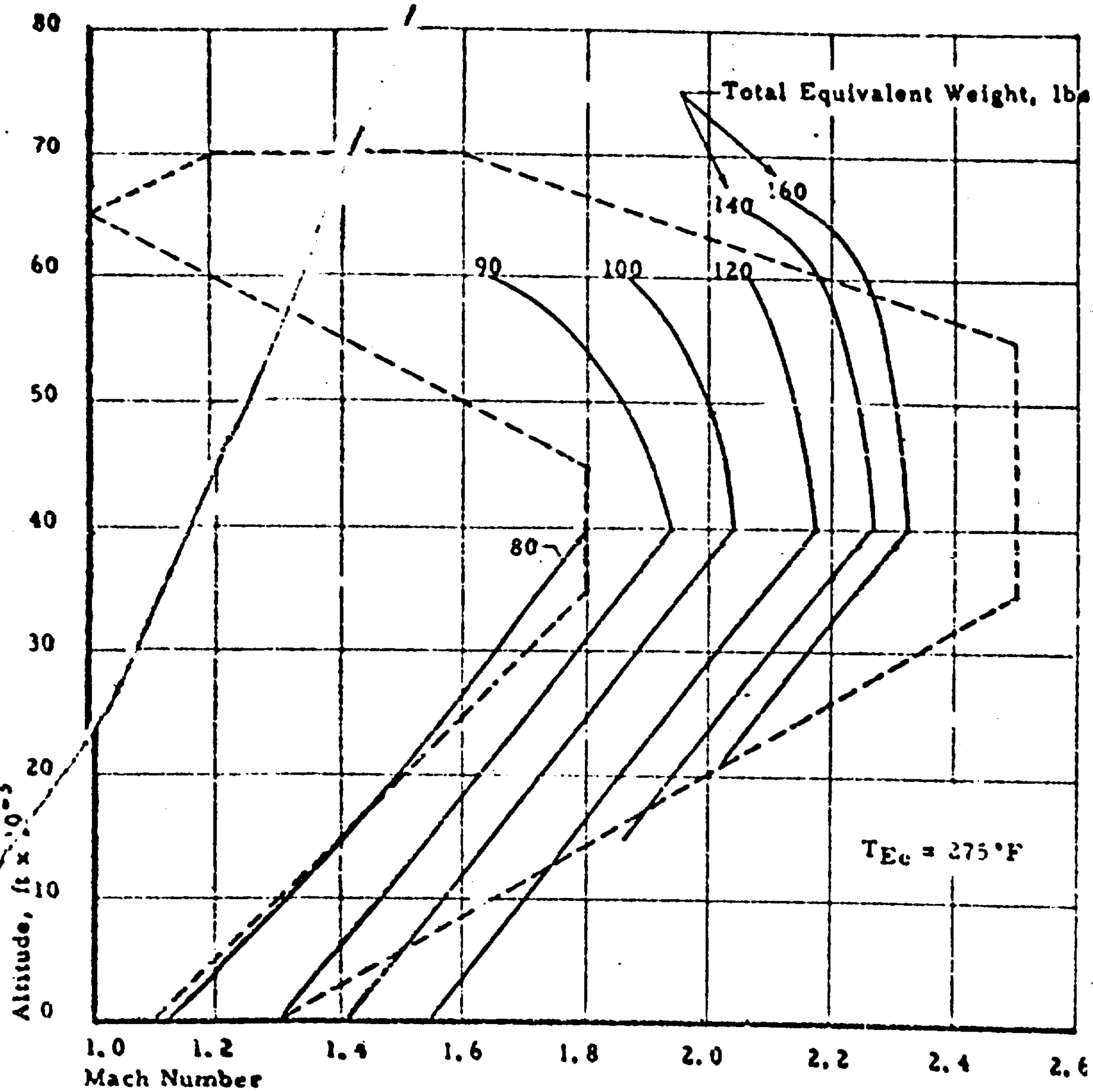


FIGURE 77 COOLING SYSTEM OPERATING LIMITS FOR A GIVEN TOTAL EQUIVALENT WEIGHT FOR A SIMPLE AIR CYCLE COOLING SYSTEM (f = 2).

3. Conclusions With Regard to Simple Air Cycle Cooling Systems

The following conclusions can be drawn from the analysis of simple air cycle cooling systems:

1) The system imposes a reasonable equivalent total weight for operation at flight speeds up to Mach 2 at 40,000 feet altitude. At higher and lower altitudes, the flight velocity for equal equivalent weight must be reduced primarily because of low air density at high altitudes and because of high ram temperatures at lower altitudes.

2) The system is very well suited to cooling equipment that presents varying temperature requirements and for that reason is not directly comparable to vapor cycle systems with an equal equipment exit temperature. It is interesting to note that for typical air cycle systems the average temperature of the air in the equipment with an exit temperature of 275°F is about 155°F, very nearly equal to the average temperature for a vapor cycle system with an equipment exit temperature of 160°F.

3) The system is inherently well adapted for cabin cooling. The same system could be used for both if of adequate capacity.

4) The total equivalent weights increase very rapidly for flight velocities above Mach 2 or for flight at altitudes above 60,000 feet, which are approximately upper limits for this type of cooling system.

5) Simple air cycle cooling systems have been highly developed and are widely used in present aircraft.

E. Regenerative Air Cycle Cooling Systems

1. Basic Considerations

The simple air cycle cooling system, considered in the preceding section, was found to impose a severe aircraft performance penalty for supersonic flight conditions. The rapid deterioration in performance of the air cycle cooling system with increasing speed, typified by figure 73, can be largely overcome through application of the principle of regeneration. The regenerative cooling system is shown schematically in figure 78. The system in its basic form does not include the precooler. The key element of the system is the regenerative heat exchanger, which serves to cool the hot compressor bleed air before it enters the expansion turbine where further cooling takes place. Since the air flowing back through the regenerator for cooling the incoming bleed air is necessarily colder than the maximum allowable equipment exit

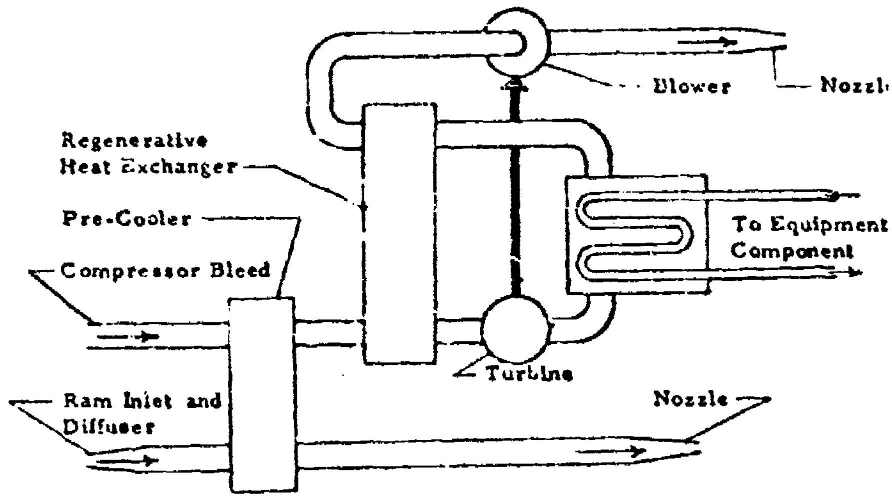


FIGURE 78 REGENERATIVE AIR CYCLE COOLING SYSTEM

temperature (T_{E_e})(considered to be from 160° to 275°F in this study), it is evident that a turbine inlet temperature approaching T_{E_e} can be achieved with the regenerative configuration. Because of the extremely high compressor bleed air temperatures encountered in supersonic flight, the basic regenerative system (which does not include the pre-cooler) requires an excessively heavy and bulky heat exchanger. This difficulty is alleviated with the modified regenerative system shown in the illustration. With this arrangement, the hot compressor bleed air is pre-cooled by ram air before entering the regenerative exchanger. This enables a considerable reduction in the size and weight of the regenerative exchanger for two reasons: first, the required effectiveness is reduced for a given turbine inlet temperature and, second, an aluminum alloy extended surface construction is feasible because of the reduced inlet temperature. Ram air required for cooling can be conveniently obtained from the engine inlet. The relatively high inlet pressure recovery possible with this arrangement permits a high degree of thrust recovery at supersonic speeds, thus keeping the drag produced by the cooling airflow to a minimum. Similarly, the penalty resulting from using compressor bleed air for indirect cooling can also be reduced by thrust recovery when the bleed air is discharged from the aircraft.

In the form shown in figure 78, the regenerative cooling system utilizes a circulating transfer fluid to convey heat rejected by the equipment being cooled to the intermediate heat exchanger, where it is carried away by the cool air leaving the turbine. An indirect cooling system of this type is well adapted to the centralized cooling system concept since the weight, size, and power requirements for the transfer system are relatively small. This is especially true of the regenerative cooling system because of the low fluid circulation rate required as a result of the low temperature of the transfer fluid leaving the intermediate exchanger (because of the small surface area of the distribution lines, heating of the transfer fluid en route to the equipment component is not significant).

Formal methods for analyzing the performance of the regenerative cooling system are presented and discussed in the next section.

The principle of regeneration can be applied to the air cycle cooling system in various ways other than the application considered in detail herein. An exhaustive study of the various embodiments of the regenerative air cycle system is beyond the scope of this report. The intent of the following analysis is to indicate the potential possibilities of the regenerative air cycle principles as applied to a specific cooling system configuration.

2. Method of Analysis and Assumptions

The analytical procedure used for determining the characteristics and aircraft performance effects of the regenerative cooling system shown in figure 78 is presented in the following pages, including generalized relationships required in the analysis. The treatment is presented in a relatively simple form with emphasis upon factors having the greatest significance in so far as the overall operation of the cooling system is concerned.

For specified values of cooling load, kw, and allowable equipment exit temperature, T_{Ee}, the required bleed airflow rate can be expressed in terms of the following factors: the bleed air temperature at the inlet to the regenerative exchanger, T_{Ri}, the total temperature drop produced by the expansion turbine, ΔT_t, and the respective effectiveness of the regenerative and intermediate heat exchangers, e_R and e_I. The individual effectivenesses are defined as follows:

Regenerative exchanger

$$e_R = \frac{T_{Ri} - T_{Re}}{T_{Ri} - T_{Ie}} \tag{120}$$

Intermediate exchanger

$$e_I = \frac{T_{Ee} - T_{Ei}}{T_{Ee} - T_{Ii}} \tag{121}$$

In these equations, T_{Ei} and T_{Ee} are the inlet and exit temperatures, respectively, of the transfer fluid flowing through the equipment component; likewise, T_{Ii} and T_{Ie} are the respective inlet and exit temperatures of the air flowing through the intermediate heat exchanger. The temperatures of the bleed air leaving the high pressure side of the regenerator are denoted by T_{Re}. By equating the air side and transfer fluid side heat transfer rates in the intermediate exchanger,

$$(wc_p)_f\,(T_{Ee} - T_{Ei}) = (wc_p)_b\,(T_{Ie} - T_{Ii}) \tag{122}$$

or

$$T_{Ie} - T_{Ii} = \frac{1}{\xi}(T_{Ee} - T_{Ei})$$

The cooling produced by the expansion turbine can be represented in the conventional form:

$$\Delta T_t = T_{ti}\, \eta_t \,(1 - r_t^{-0.286}) \qquad (123)$$

Using equation (123), the inlet temperature to the intermediate exchanger is

$$T_{Ii} = \lambda T_{Re} \qquad (124)$$

The required transfer fluid flow rate is obtained by equating the rate of heat absorption by the transfer fluid to the equipment heat rejection rate:

$$(wc_p)_l = \frac{0.95\ kw}{T_{Ee} - T_{Ei}} \qquad (125)$$

By making use of the relationships expressed by equations (120) to (125), the required air flow rate is found to be

$$w_b = \left[\frac{\dfrac{\zeta}{e_T} + \dfrac{e_R\lambda}{1 - e_R}}{T_{Ee} - \dfrac{(1 - e_R)\lambda T_{Ri}}{1 - e_R\lambda}}\right] 3.95kw \qquad (126)$$

The bleed air temperature at the regenerator inlet T_{Ri} is defined by

$$e_b = \frac{T_b - T_{Ri}}{T_b - T_T} \qquad (127)$$

where e_b is the effectiveness of the ram exchanger, T_b is the compressor bleed temperature, and T_T is the ram air temperature. It should be noted that absolute temperatures must be used in equation (126). The equipment inlet temperature T_{Ei}, in terms of the airflow rate w_b given by equation (126), is

$$T_{Ei} = T_{Ee} - \frac{3.95\,\zeta\ kw}{w_b} \qquad (128)$$

Because of the considerable aircraft performance penalty associated with bleeding air from the compressor of a turbojet engine, the bleed airflow rate must be kept as low as possible. By inspection of equation (126), it is apparent that this can be done in one or more of the following ways (considering kw and T_{Ee} fixed):

1) Increase e_R to the highest practical value

2) Decrease T_{Ri} by using a ram exchanger of high effectiveness

3) Increase by operating the turbine at a high pressure ratio

The first two means of reducing the bleed airflow rate must take cognizance of the rapid increase in heat exchanger weight with increasing effectiveness. This is particularly true of the regenerative heat exchanger. Of course, T_{Ri} will always exceed the ram air temperature T_T. The extent to which the bleed airflow rate can be reduced by increasing is limited by reduced turbine efficiency at high turbine pressure ratios. Another factor which might conceivably limit the turbine temperature drop is the possibility of icing caused by subfreezing turbine discharge temperatures. In general, a turbine pressure ratio of from 4 to 6 at a turbine efficiency of 80-85% would be a good choice. Control of the turbine pressure ratio is discussed in a subsequent part of this section.

The assumed engine operating conditions for cruising flight listed in the general discussion of air cycles are sufficient to define the compressor bleed temperature and pressure, as well as the engine inlet pressure, as a function of Mach number and altitude. The results are given in figures 66, 67, and 64 respectively.

Extraction of compressor discharge air occasions a loss of engine thrust and an increase in specific fuel consumption, both of which have a deleterious effect upon aircraft performance. The overall thrust loss is made up of two additive effects: a momentum drag due to taking the bleed air aboard the aircraft via the engine inlet and a direct thrust loss caused by the energy expended in compressing the bleed air in the engine. The latter factor is accompanied by an increase in specific fuel consumption. The equivalent weight due to compressor air extraction can be expressed in terms of the effects of increased drag (treating a thrust loss as an increment in drag) and an increased specific fuel consumption. These effects were examined analytically by employing the procedure of reference 18. The effect of air bleed can be expressed by equation (111).

As indicated by figure 78, three heat exchangers are required for the modified regenerative cooling system: the regenerative exchanger, the ram pre-cooler, and the intermediate liquid-to-air heat exchanger. Each of these heat exchangers is characterized by its effectiveness, pressure losses, size, and weight. In general, for the specified design conditions the size and weight of a heat exchanger are dependent upon: the flow rate of cooled and coolant fluids, the physical arrangement of the exchanger and characteristics of the core, the effectiveness and the permissible pressure losses. General relationships among these factors are developed in appendix I for the various types of exchangers required for air cycle systems. Particular results pertaining to the regenerative cooling system analysis are discussed below.

The major requirements of the regenerative exchanger are that it be as light and compact as possible, while still having a sufficiently high effectiveness to cool the high pressure compressor bleed air to a temperature approaching the equipment exit temperature, T_{Ee}. In as much as the ratio of coolant to cooled fluid flow rates is unit for the regenerator, a relatively large heat transfer surface area is required, especially for high effectiveness, leading to a comparatively large and heavy exchanger. Weight and size of the regenerator can be minimized through use of highly efficient extended surface cores fabricated of aluminum alloy. Because of the temperature limitation of an aluminum core, it is necessary to pre-cool the incoming bleed air in a ram air exchanger as discussed previously. This has the additional advantage, in so far as the regenerator is concerned, of enabling a reduction of effectiveness for a given exit temperature which also contributes to a saving in size and weight.

The detailed analysis of the regenerative exchanger is given in appendix I based upon heat transfer and pressure drop characteristics for an efficient aluminum extended-surface core (references 20 and 22 The results of this analysis are presented graphically in figure 58 for the following assumed conditions: (1) triple-pass, counter-crossflow configuration, (2) length-width ratio $l_a / l_c = 2.0$ (for a single pass of the cooled airflow, see figure 59). Figure 62 gives the regenerator weight in pounds per pound/second of bleed airflow as a function of the exchanger effectiveness e_R defined by equation (120) and a pressure dorp parameter defined by the expression

$$\phi_{Rb} = \left(\frac{\Delta P}{P_i}\right)_b \frac{(\delta_i)_b^2}{\theta_b} \qquad (129)$$

In the above equation, the subscripts i and b refer, respectively, to inlet conditions and to the cooled flow; thus $(\delta_i)_b$ refers to the inlet pressure of the high pressure bleed air entering the exchanger while θ_b relates to the mean temperature of the high pressure flow. For the low pressure (cooling flow) side of the regenerator, the pressure loss is defined, in the general case, by the expression:

$$\phi_{R2} = \left(\frac{\Delta P}{P_i}\right)_2 \frac{(\delta_i)_2^2}{\theta_2} = \left(\frac{l_c}{l_s}\right)^{2.317} \phi_{Rb} \qquad (130)$$

where the subscript 2 refers to the coolant flow. For $l_s / l_c = 2.0$, corresponding to the conditions assumed for figure 62,

$$\phi_{R2} = 0.162\ \phi_{Rb} \qquad (131)$$

Figure 62 indicates the rapid increase in regenerator weight with increasing effectiveness and also shows how the weight decreases when the exchanger pressure losses are permitted to increase for a given effectiveness and flow rate. The volume of the regenerator exchanger is

$$V_R = 0.0368\ W_R \qquad (132)$$

The relationships presented in figure 62 and equations (129) and (130) must be considered to apply to a "design" operating condition for the cooling system, i.e., for specified design values of e_R, w_b, and ϕ_b (or for an equivalent condition defining pressure losses). For "off-design" operating conditions, the regenerator effectiveness and pressure losses will of course be different. As will be shown subsequently, however, the bleed airflow rate remains substantially constant with the regenerative system over a wide range of flight conditions.

The function of the ram exchanger, or pre-cooler, is to cool the very hot compressor bleed air prior to its entry into the aluminum regenerator so as not to exceed the temperature limitations of the latter. The required pre-cooler effectiveness is found from equation (127) in terms of the compressor bleed air temperature, the ram air temperature, and the desired regenerator inlet temperature. Because of the high bleed air temperatures to be expected at high speeds (1260°R or higher), a tubular stainless steel core is appropriate for the ram exchanger, although a copper alloy extended-surface core could also be used. In all probability, the former would have the advantage of lighter weight whereas the latter would be more compact. In the present study, the stainless steel tubular construction was assumed. Representative core characteristics were obtained from references 19 and 20. The weight of the ram exchanger is shown in figure 61 as a function of the bleed flow rate w_b, effectiveness e_b, and the ram air pressure loss parameter ϕ_{ex-a}, defined by

$$\phi_{ex-a} = \left(\frac{\Delta P}{P_i}\right)_{ex-a} \frac{(\delta_i)^2_{ex-a}}{\theta_{ex-a}} \qquad (133)$$

Figure 61 is based upon the detailed analysis given in appendix I together with the following assumed conditions: (1) triple-pass counter-crossflow configuration, (2) $l_a / l_c = 3.0$, and (3) ratio of coolant to cooled flow rate, $w_a/w_b = 1.5$. The volume of the ram air exchanger is given by

$$V_{ex} = 0.048\ W_{ex} \qquad (134)$$

The purpose of figure 61 is to define the required size and weight of the ram exchanger for specified design values of effectiveness, flow ratio, and allowable pressure drop; off-design conditions can be analyzed by using relationships given in appendix

The intermediate heat exchanger serves to cool the transfer fluid with cool air leaving the expansion turbine (see figure 7?). In order to keep the required bleed airflow rate to a minimum, it is apparent from examination of equation (126) that $\zeta = (wc_p)_b/(wc_p)$ should be as small as possible and e_I as high as possible. A suitable core arrangement for the liquid to air intermediate exchanger is shown in figure 33. The core itself consists of a large number of closely spaced finned tubes through which the transfer fluid flows in series. An analytical treatment of the intermediate exchanger is given in appendix I based upon air side heat transfer and pressure drop data from reference 19. Results of the analysis are plotted in figure 79 for the assumed conditions: $e_I = 0.8$, $\zeta = 1.0$ and 1.5. The wet weight of the exchanger is based upon the core being filled with water. As in the case of the previous exchangers, the weight and size of the intermediate heat exchanger is given in terms of the air side pressure drop parameter

$$\phi_I = \left(\frac{\Delta P}{P_i}\right) \frac{(\delta_i)^2_I}{\theta_I} \qquad (135)$$

The volume occupied by the intermediate exchanger is

$$V_I = 0.05\ W_I \qquad (136)$$

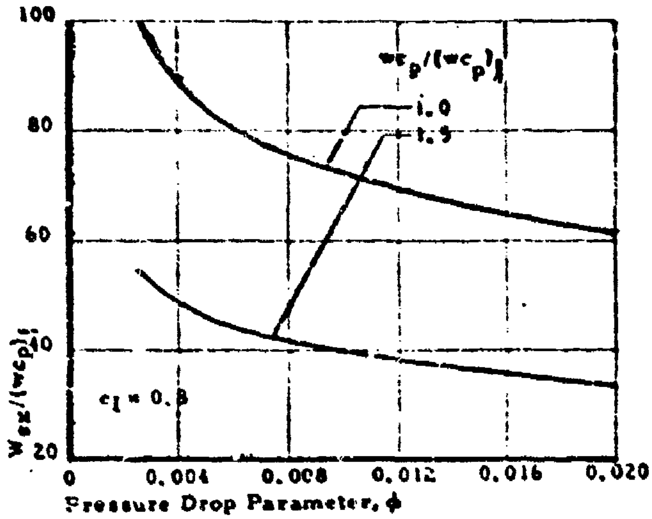


FIGURE 79 WET WEIGHT OF A LIQUID TO AIR HEAT EXCHANGER

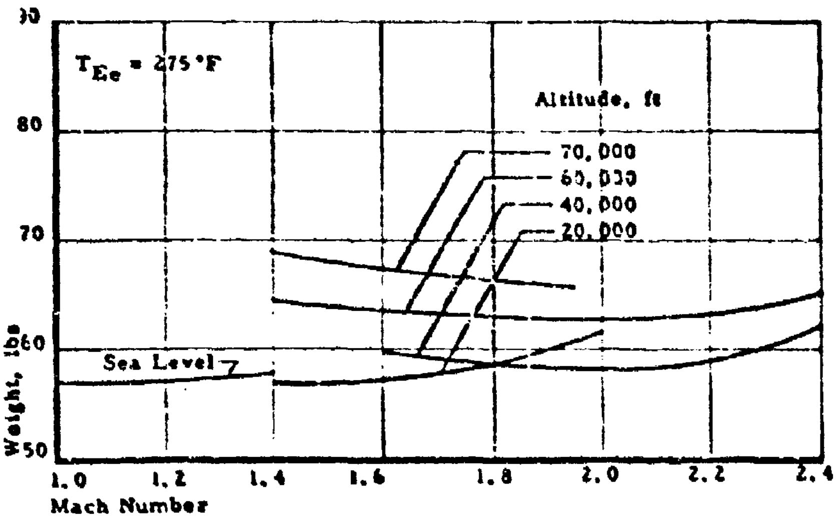


FIGURE 80 DEAD WEIGHT FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM

Ram air cooling of the compressor bleed air may result in appreciable momentum drag at high speeds. The net momentum drag can be minimized through thrust recovery, achieved by exhausting the ram air through a convergent nozzle. Effective thrust recovery is promoted by the following factors: (1) efficient inlet pressure recovery, (2) low pressure losses in ducting and ram exchanger, and (3) addition of heat to the ram air. The first of these desired conditions can be best realized by obtaining the ram air supply from the engine inlet. Pressure losses can be kept to a minimum by proper selection of heat exchanger and duct geometry, while the third factor mentioned above is automatically satisfied, becoming a significant factor for high e_R and low ζ.

The size and weight of the required ducting can be best defined on the basis of allowable duct pressure losses as given by equations (101), (102), and (103). The numerical coefficient F_{dt} in equations (101) and (102) is $F_{dt} = 2.5$. This will approximately account for the increased friction factor in the inlet region of the duct as well as the diffuser losses at the entrance to the ram exchanger. The duct diameter d is found from equation (102) in terms of the known flow rate w_a, duct length ℓ, inlet conditions θ_T and δ_T, and the allowable pressure drop ratio $(\Delta p/p_i)_{dt}$. (δ_T is found from figure 64.) The above procedure affords a reasonable basis for estimating the required inlet duct size and weight. The exit ducting can be treated in a similar way.

The momentum drag resulting from the ram cooling flow can be expressed by equation (97). The nozzle pressure ratio is easily determined in terms of the inlet pressure ratio (figure 64) and the ducting and heat exchanger pressure losses.

Part of the aircraft performance penalty resulting from compressor bleed can be offset by thrust recovery when the bleed air is ejected from the aircraft. The thrust recovery can be expressed by equation (97). Thrust recovery clearly depends upon conservation of pressure through the bleed air passages, i.e., the ram exchanger, ducting, the regenerative exchanger, expansion turbine, intermediate exchanger, and, finally, the exit ducting. The individual pressure losses may be evaluated on the basis of the pressure loss considerations previously discussed, thus enabling the determination of the pressure level at the exit from the low pressure side of the regenerative exchanger. This pressure level is increased by the compressor located in the exit duct which serves as a load for the expansion turbine (see figure 78). In terms of the compressor adiabatic efficiency η_C, the pressure ratio developed by the compressor can be expressed by equation (98).

The degree of thrust recovery possible with the regenerative cooling system is to a great extent dependent upon the means provided for controlling the bleed airflow. If maximum thrust recovery is to be obtained over a range of flight conditions, control must be exercised over both the turbine nozzle area and the exit nozzle area. A variable-area turbine, such as that described in reference 27, enables control of the bleed airflow rate over a rather wide range of bleed pressures, whereas control of the turbine pressure ratio necessitates a variable-area exit nozzle. From the standpoint of practicability, it would appear desirable to employ a fixed-area exit nozzle providing effective thrust recovery for the design cruise condition, together with a variable-area turbine for control of the bleed airflow rate for all flight conditions. This approach avoids difficult flow control problems at the expense of less efficient cooling system performance for conditions other than the design cruising condition. Subsequent calculations indicate that the regenerative cooling system imposes a relatively moderate aircraft performance penalty even when bleed air thrus recovery is ignored.

3. Results of Regenerative Air Cycle Cooling System Analysis

The analytical procedure presented in the foregoing section enables the determination of the pysical characteristics and the aircraft performance penalty of the regenerative air cycle cooling system for specified conditions. The independent variables involved in the analysis are as follows: cooling load (kw), equipment exit temperature (T_{Ee}), Mach number (M), and altitude (H); also duct and transfer line lengths. Additional variables required for the analysis and subject to choice are the following: effectiveness of ram, regenerative, and intermediate heat exchangers (e_b, e_R, and e_I, respectively), turbine pressure ratio (r_t), and regenerator inlet temperature (T_{Ri}); in addition, the heat exchanger and ducting pressure losses may be treated as design variables which can be chosen to meet the overall requirements of the cooling system.

The fundamental point of view adopted for the numerical analysis is to interpret each Mach number-altitude combination of figure 1 as a design cruising condition and to determine the cooling system characteristics and performance for this condition, thus involving no off-design operation. An alternate procedure would be to select a certain design flight condition, at which the cooling system geometry is fixed, and then to determine the performance of this fixed cooling system over a range of altitudes and speeds. Although the latter approach is more representative of actual operating conditions of a cooling system, it is rather impractical for the present purpose mainly because detailed engine operating conditions are not available. As a matter of fact, the subsequent analysis indicates that the regenerative cooling system performance is not subject to great changes with changing flight conditions.

Another basic factor to be considered is that of optimization of the cooling system. Optimization (in the strict sense of the term) was not attempted for the regenerative cooling system. The system was analyzed with an engineering optimization as discussed in section III of this report.

The bulk of the numerical calculations were carried out for a cooling load of kw = 10 kilowatts with T_{Ee} = 275°F. The effects of changing kw and T_{Ee} are discussed later. Design variables such as heat exchanger effectiveness, pressure losses, etc., are given below. It will be noted that the analysis is based upon a constant regenerator inlet temperature T_{Ri} = 1000°R, independent of Mach number and altitude. Reasons for this choice are indicated in the discussion.

Assumed Design Variables for Analysis of the Regenerative Cooling System

$$kw \quad 10 \text{ at } T_{Ee} = 735\,°R$$
$$T_{Ri} = 1000\,°R$$

Heat Exchanger Characteristics

$$e_R = 0.8 \text{ at } \phi_{R-b} = 0.01$$
$$e_I = 0.8 \text{ at } \phi_I = 0.002 \text{ and } \zeta = 1.0$$
$$e_b = \frac{T_b - T_{Ri}}{T_b - T_r} \text{ at } \zeta = 1.5$$

(The choice of $_a$ is discussed below.)

Air Cycle Machine Characteristics

Radial turbine with variable-area nozzle:
$r_t = 5.0$ at $\eta_t = 0.8$
Radial compressor driven by turbine:
$\eta_C = 0.2$

Ducting Characteristics

Bleed air duct - engine to regenerator:
$(\Delta p)/p_b = 0.02$, $l_{dt} = 20$ ft
Bleed air exit duct:
$(\Delta p)/p_t = 0.02$, $l_{dt} = 6$ ft
Ram cooling duct:
$(\Delta p)/p_T = 0.04$, $l_{dt} = 16$ ft
(based on $(\Delta p)/p_i = 0.02$ for inlet and exit ducts)

Nozzle Efficiency

For both the ram cooling duct and the bleed air exit duct, exhaust nozzle efficiencies of $\eta_n = 0.08$ are assumed, based on air discharge approximately 30° off the flight direction.

The data given in table 1 enable a complete analysis of the cooling system characteristics and performance over the Mach number-altitude range shown in figure 1 for kw = 10 and $T_{Ee} = 275$ °F. The analysis is based upon the methods presented in part B. The assumed regenerator operation conditions result in a constant bleed airflow rate, w_b, independent of altitude and Mach number. This in turn leads to constant values of regenerator weight, W_R, and intermediate exchanger weight, W_I, as well as a constant weight for the air cycle machine, W_t. These values are as follows:

Bleed airflow rate:

$w_b = 0.27$ lb/sec (equations 5 and 8)

Regenerator weight and volume:

$W_R = 26.8$ lb (figure 62 and w_s)
$V_R = 0.985$ ft^3 (equation 16 and W_R)

Intermediate exchanger weight and volume:

$W_I = 6.8$ lb (figure 79 and w_s)
$V_I = 0.34$ ft^3 (equation 20 and W_I)

Air cycle machine weight:

$W_t = 8.0$ lb (equation 93)

Remaining cooling system characteristics such as duct sizes and weights, pre-cooler size and weight, momentum drag due to cooling air, and bleed air thrust recovery are dependent upon Mach number and altitude. Some of the more important of these characteristics, which were determined by applying the analytical methods discussed previously are illustrated in figures 79 to 82.

The equivalent total weight of the regenerative cooling system is shown in figures 83 and 84 as a function of altitude and Mach number for the specified conditions. Results are given in figure 83 for zero bleed air thrust recovery and in figure 84 for the thrust recovery possible under the conditions listed.

The operating limits for regenerative air cycle cooling systems of different total equivalent weights are indicated on the altitude-Mach number envelope in figure 85 assuming zero thrust recovery and in figure 86 assuming a thrust recovery as indicated in the analysis.

4. Effects of Changes in the Regenerative Air Cycle Cooling System Operating Conditions

The effects of departures from the assumed cooling system operating conditions used in the preceding analysis are discussed below for some of the more important cooling system variables.

1) The effective weight penalty is, for all practical purposes, directly proportional to the cooling load. To a great extent, other cooling system characteristics (such as heat exchanger bulk, momentum drag losses due to ram cooling, etc.) are also proportional to cooling load.

2) The effect of changing T_{Ee} from the value T_{Ee} used in this analysis is illustrated in figure 83 for M = 2.0 at 60,000 feet altitude. It is considered very unlikely that equipment cooling conditions would ever dictate a value of T_{Ee} less than 275 °F for the regenerative cooling system because of the low coolant temperature at the equipment inlet (about 86 °F for the conditions given).

3) The preceding calculations were based upon compressor bleed temperatures given in figure 66, which is based upon a maximum compressor discharge temperature of 1250 °R. The effect of changes in T_b upon the overall effective cooling system weight is shown in figure 87 for M = 2.0 and 2.4 at 60,000 feet altitude. For T_{Ri} = 1000 °R, the ram exchanger effectiveness changes with T_b as indicated by equation (127) maintaining the same $_a$ used for T_b = 1250 °R. Figure 87 indicates that

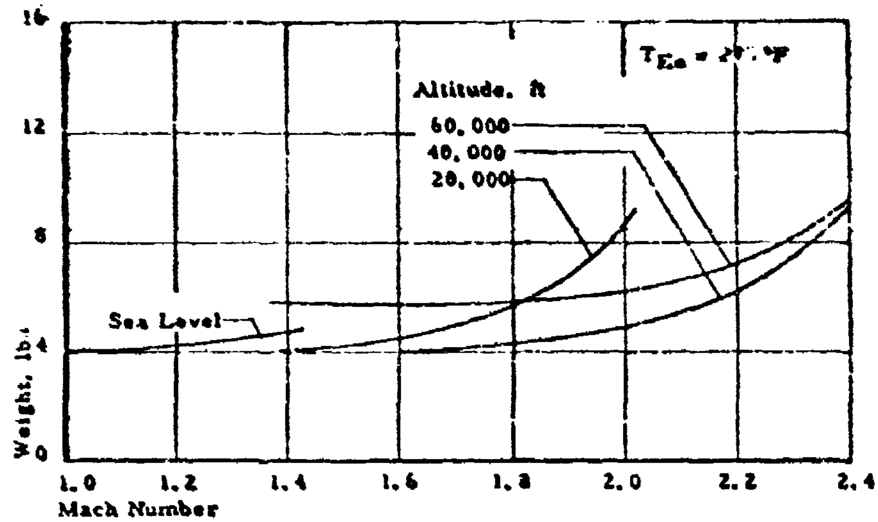


FIGURE 81 RAM AIR HEAT EXCHANGER WEIGHT FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM

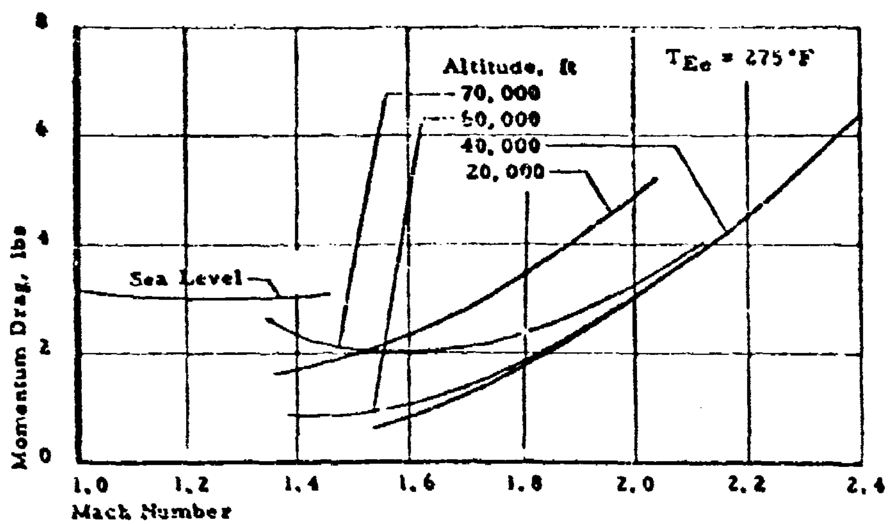


FIGURE 82 RAM AIR MOMENTUM DRAG FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM

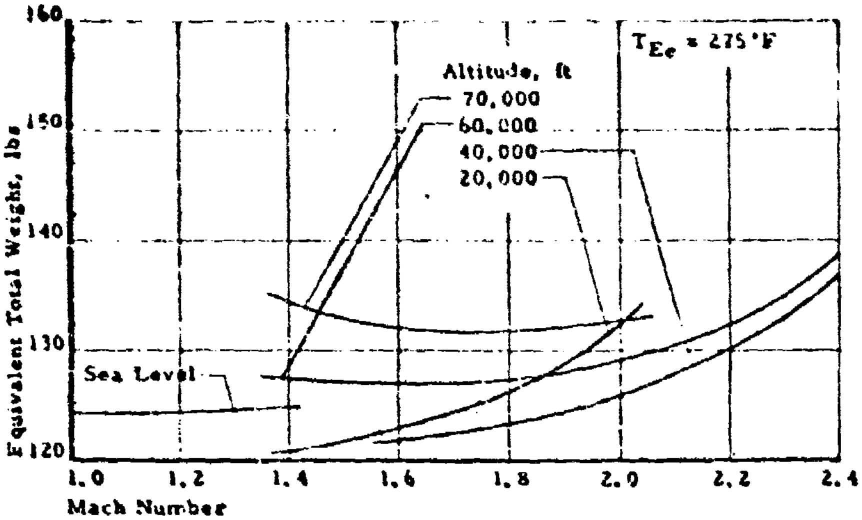


FIGURE 83 EQUIVALENT TOTAL WEIGHT FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM WITH OUT BLEED AIR THRUST RECOVERY

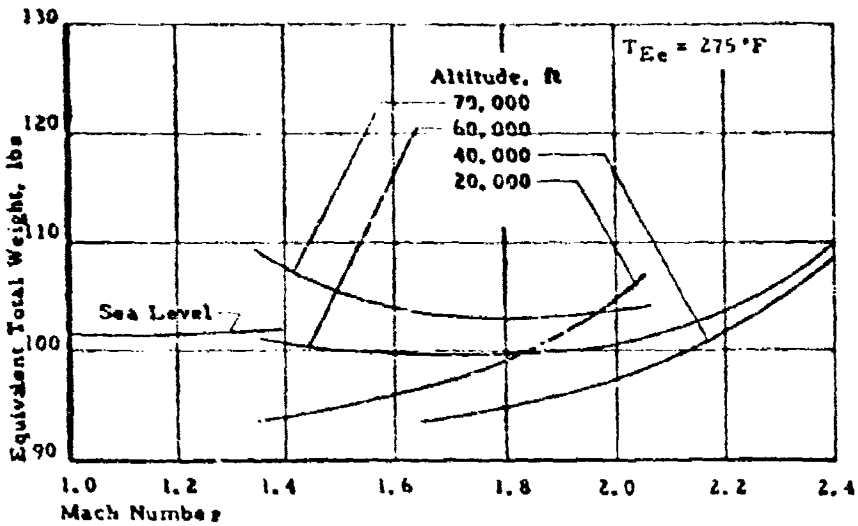


FIGURE 84 EQUIVALENT TOTAL WEIGHT FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM WITH BLEED AIR THRUST RECOVERY

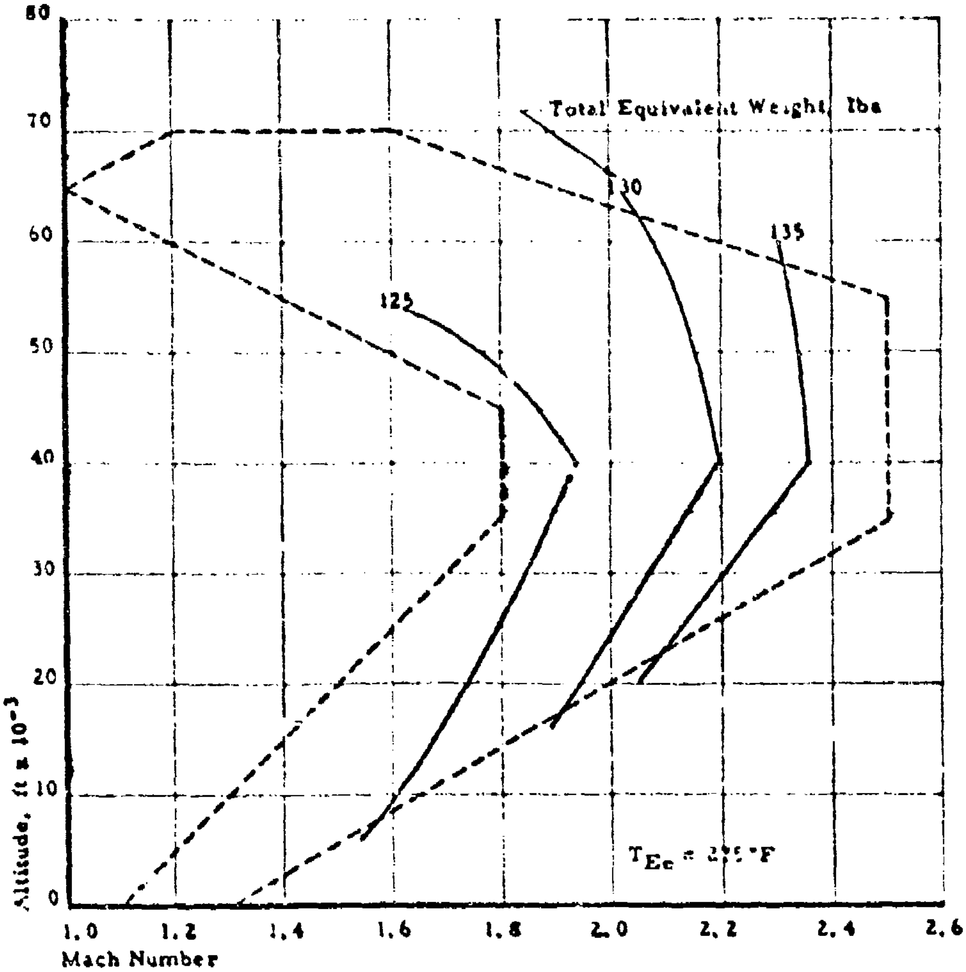


FIGURE 85 COOLING SYSTEM OPERATING LIMITS FOR A GIVEN TOTAL EQUIVALENT WEIGHT FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM WITH OUT BLEED AIR THRUST RECOVERY

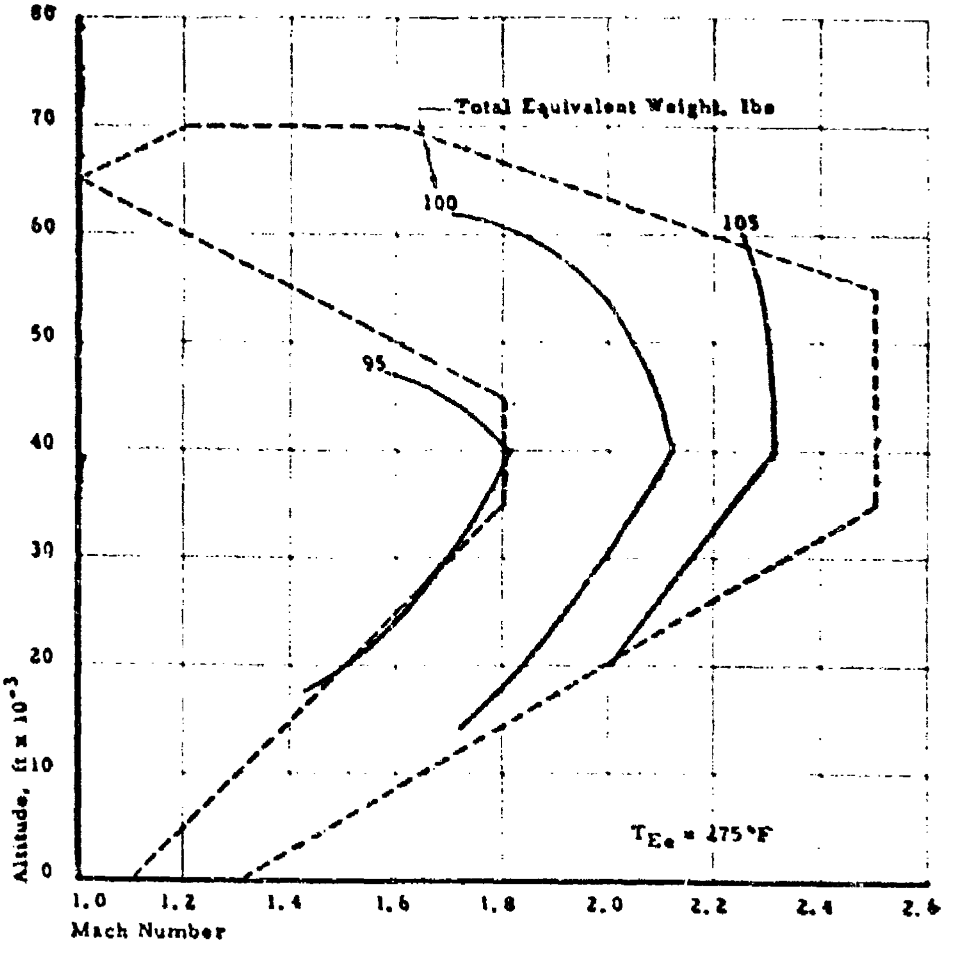


FIGURE 86 COOLING SYSTEM OPERATING LIMITS FOR A GIVEN TOTAL EQUIVALENT WEIGHT FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM WITH BLEED AIR THRUST RECOVERY

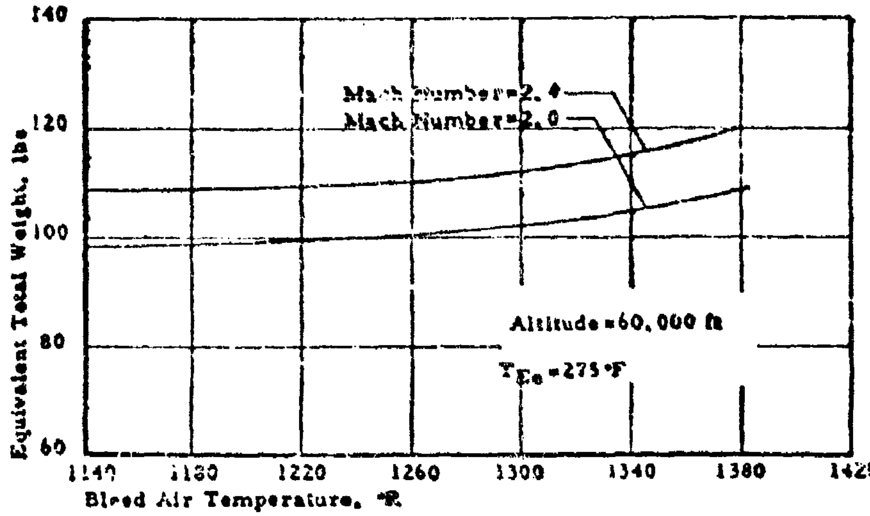


FIGURE 87 EQUIVALENT TOTAL WEIGHT VERSUS COMPRESSOR BLEED AIR TEMPERATURE FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM

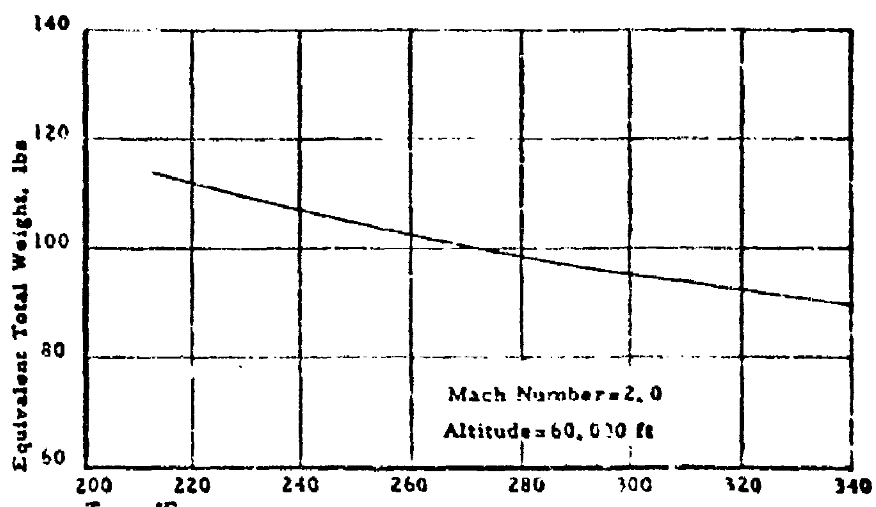


FIGURE 88 EQUIVALENT TOTAL WEIGHT VERSUS EQUIPMENT EXIT TEMPERATURE FOR A REGENERATIVE AIR CYCLE COOLING SYSTEM

the effect of T_b in the range from 1150° to 1350°R is small for both M = 2.0 and 2.4.

5. Conclusions With Regard to Regenerative Air Cycle Cooling Systems

The following conclusions can be drawn from the foregoing analysis of the regenerative air cycle cooling system:

1) The regenerative cooling system imposes a relatively small aircraft performance penalty, particularly in the case of bleed air thrust recovery where the maximum cooling system effective weight penalty is about 120 pounds (for M = 2.5 at 4,000 feet altitude) for a cooling load of 10 kilowatts at T_{Ee} = 275°F. Without bleed air thrust recovery, the effective weight penalty for these conditions is about 150 pounds.

2) The equipment inlet temperature attained with the regenerative cooling system is relatively low even at high speeds, values of from 60° to 90°F being easily attained. This is beneficial for equipment cooling because of the capability of cooling equipment items having a range of operating-temperature limits in a given equipment package; also, fluid circulation rates can be relatively low, which results in a saving in weight and power requirements for the transfer fluid system. A further advantage of a low coolant temperature is the potential application of the cooling system to cabin cooling as well as equipment cooling; thus, the coolant could first be passed through a heat exchanger in the cabin for cooling the cabin air, after which the coolant could be employed in the usual way for equipment cooling. A relatively small coolant temperature rise would occur in cooling the cabin (assuming reasonably good insulation).

3) The regenerative air cycle system could be designed so as to be quite compact and light in weight.

4) Ground cooling can be achieved with the engine idling.

5) Because of the probability that large compressor bleed and engine inlet airflows will be required for cooling of the engine itself, the use of an efficient air cycle cooling system, with its relatively small demands on compressed air, becomes relatively more desirable and more readily integrated into the overall aircraft design than would be the case if special provisions were required for bleeding compressor or inlet air for the cooling system alone.

6) The regenerative cooling system does not appear to depend upon developments in heat exchangers or air cycle machines beyond the present state of the art for effective application to equipment or cabin cooling.

SECTION VII

EXPENDABLE COOLANT EQUIPMENT COOLING SYSTEMS

A. Basic Considerations

Aircraft that have mission requirements of relatively short duration or for which the high speed portion of the flight will be limited to relatively short dashes either by mis   or by fuel considerations can efficiently utilize expendable coolant equipment cooling systems. This concept of cooling involves a fluid, carried on board the aircraft, that absorbs heat (usually by changing state) while cooling the equipment and is then dumped overboard in a higher energy state.

The prime requirement is for a substance that can absorb a large amount of heat per pound at the required temperature level. Since the coolant is expendable, the weight will be a direct function of the heat absorption capabilities of the coolant. Other considerations are physical properties such as freezing point, vapor pressure, toxicity, and corrosivity and practical considerations such as availability, cost, etc.

The expendable coolant could be any fluid or solid that can absorb heat during a change of state or it could be a substance that would undergo an endothermic chemical reaction. The most promising expendable coolants, investigated during the course of this study, are fluids which change in state from a liquid to a vapor, a process during which the fluid absorbs the latent heat of vaporization.

The utilization of the heat of fusion does not appear practical because the energy change is usually much less during melting than during vaporization and, further, solids are not readily transported through a system. No endothermic chemical reactions that appeared practical enough to warrant further investigation were noted in a search through references 10 and 11. Consequently, further study of expendable coolant systems was limited to fluids that would change state from liquid to vapor at suitable pressures and temperatures.

Fluids that freeze within the temperature range of interest, notably water, have been considered for applications in systems designed so that freezing would not render the system inoperable. However, in evaluation calculations, the heat of fusion has not been considered as it is available only when the initial temperature is below the freezing point.

In the selection of an expendable coolant, the boiling points of the fluid at the applicable pressures is a prime consideration. In fact, this is the property that defines the temperature level that can be maintained with the fluid. The boiling point must be below the equipment temperatures if cooling is to be achieved. Thus at sea level water boils at 212°F, ethyl alcohol at 173°F, methyl alcohol at 148.5°F, and ammonia at -28.5°F. The above fluids therefore cannot be used to cool below the indicated temperatures at sea level. The boiling point is that temperature at which the vapor pressure is equal to the atmospheric pressure; therefore, the boiling point will decrease as the ambient pressure decreases. A particular fluid is therefore capable of cooling equipment to a lower temperature level at altitudes above sea level. Curves indicating the variation of boiling point with altitude for several fluids are shown in figure 89. The water-ethylene glycol, water-ethyl alcohol, and water-methyl alcohol solutions are each of proportions that freeze at -65°F.

The vapor pressure is also significant in so far as storage is concerned. If the fluid is stored on the aircraft at ambient pressure, evaporation will result in a temperature such that the vapor pressure is equal to the ambient pressure. Evaporation of the fluid must compensate for heat flow into the fluid from the surroundings. If the fluid container is pressurized, the vapor pressure at the storage temperature determines the pressure that applies.

The amount of cooling that can be secured by the vaporization of a liquid (vaporizing at T_V and stored at T_s) is given by the equation (assuming constant specific heats)

$$Q = L_V - c_{p\ell} \, (T_s - T_V) \tag{137}$$

The latent heat decreases with increasing temperature. Assuming constant specific heats, the latent heat (L_V) at a temperature T_V, in terms of the specific heats and the latent heat (L_o) at a temperature T_o is given by the equation

$$L_V = L_o - (c_{p\ell} - c_{pg}) \, (T_V - T_o) \tag{138}$$

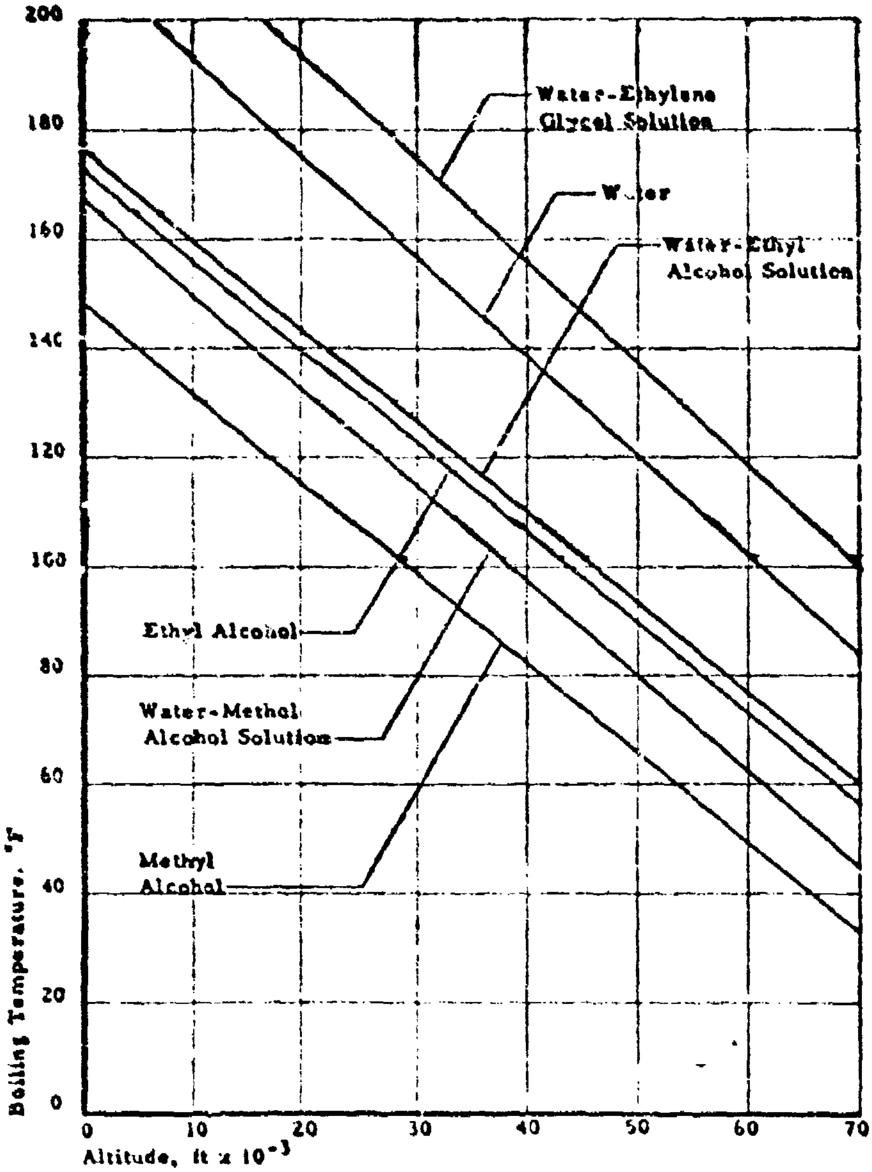


FIGURE 89 BOILING TEMPERATURE VERSUS ALTITUDE FOR SEVERAL EXPENDABLE FLUIDS

The above equations indicate that even though the latent heat decreases as the vaporizing temperature increases, the net cooling effect increases as the vaporizing temperature increases. This is because the sensible heat of the stored liquid reduces the net cooling effect when the storage temperature is above the evaporating temperature. In the event that the storage temperature is below the vaporizing temperature, the sensible heat will increase the net cooling and again a high vaporizing temperature is indicated for maximum cooling.

The cooling effect can also be expressed in terms of the latent heat of vaporization at the liquid storage temperature T_s by substituting equation (138) (with proper subscripts) into equation (137).

$$Q = L_s - c_{pg} (T_s - T_V) \tag{139}$$

Equation (137) indicates that, when the storage temperature is higher than the vaporizing temperature, the net cooling effect is less than the latent heat at the storage temperature. The reduction is equal to the product of the specific heat of the gas and the difference between the storage and vaporizing temperature. Equation (139) bears out the observation that maximum net cooling is obtained at a maximum evaporating temperature. The above analysis indicates the importance of considering the latent heat at the storage or the vaporizing temperature, the specific heat of the gas or the vapor, respectively, and the temperature differences between the storage container and the evaporator. Only with the above considerations can a valid comparison of various expendable coolants be made.

The freezing point and boiling point of a fluid defines the range of applicability for use as an expendable coolant. The latent heat of vaporization defines the cooling effect. Other fluid characteristics and properties may or may not be significant for a particular application. The general characteristics pertaining to expendable coolant applications for some of the more promising fluids are discussed below.

Out of a large number of fluids considered, six have been selected as possessing sufficient merit to warrant a thorough analysis of applicability and of system characteristics. The six fluids are water, water-methyl alcohol solution, methyl alcohol, water-ethyl alcohol solution, ethyl alcohol, and ammonia. Some of the more significant thermal properties are listed in table 4.

TABLE 4 PROPERTIES OF EXPENDABLE COOLANTS

	Vapor Pressure (psia)			Latent Heat (Btu/lb)		Freeze Temperature (°F)	Boiling Temperature (°F)	
	100°F	150°F	200°F	100°F	200°F		Sea Level	40,000 ft
Water	0.949	3.718	11.526	1037	978	32	212	137.6
Ammonia	211.9			478	350*	-147	-28.5	
Water-Ethyl Alcohol Solution	2.05	8.04	23.8	584	552	-65	176	110
Water-Methyl Alcohol Solution	3.00	9.99	29.0	590	560	-65	167	96
Water-Ethylene Glycol Solution	0.62	2.42	7.3	386	364	-65	232	154
Ethyl Alcohol	2.24	8.64	25.8	395*	355*	-238	173	106
Methyl Alcohol	4.56	15	40.2	490*	450*	-208	148.5	82

*Estimated Approximate Values
Ethyl Alcohol 367 at 173°F
Methyl Alcohol 473 at 148.5°F
Ammonia 588 at -28.5°F, 449 at 125°F

Among the fluids considered as expendable coolants, the thermodynamic properties of water are unique. In fact, the desirable properties are so outstanding that considerable work and design effort is justified to circumvent the one or two disadvantages encountered in its use. The only notable liabilities of water are its relatively high freezing and boiling points. The freezing point problem can be eliminated by the use of a heat transport fluid. The high boiling point requires heat exchange at small temperature differences and equipment that can operate at the necessary temperature. An indication of the temperature at which cooling can be secured is indicated by the boiling point and its variation with altitude. The decrease in boiling point is very nearly linear with altitude from 212°F at sea level to approximately 100°F at 65,000 feet altitude. Figure 89 illustrates the variation of the boiling point with altitude for water and for several other fluids. The curves of figure 89 are linear approximations but are within about two degrees for the entire range covered

The latent heat of vaporization of water is more than double that of any of the other fluids (except water solutions). The latent heats at significant temperatures are listed in table 4. The safety, availability, non-inflammability, and non-toxicity of water are of course unequaled.

A primary advantage of ammonia, for some applications, is its low boiling point, -28°F at standard sea level pressure. The latent heat is relatively high, 478 Btu/lb at 100°F. The latent heat of 598 Btu/lb at -28°F is not significant for cases where the fluid is pressurized and stored at a higher temperature. Storage at rather high pressure is necessary to prevent evaporation. Ammonia is highly toxic, involves rather severe handling problems, and possesses undesirable chemical properties. Ammonia is worthy of consideration only for cases where a very low temperature is needed.

The addition of methyl or ethyl alcohol to water to lower the freezing point is a means of utilizing water for some applications where freezing cannot be tolerated. The solution does have a somewhat lower boiling point as indicated by the curves of figure 89. However, it should be pointed out that these solutions (freezing at -65°F) are not azeotropic and therefore the solutions do not have a true fixed boiling point; the composition of the vapor will be slightly different from the liquid solution resulting in a solution that will vary in composition and the boiling point will gradually increase as the liquid boils off. Such solutions can be used for cooling if the pressure is such that water will boil, or even at the lower boiling temperature for a short time, e.g., while climbing to altitude. The addition of the alcohol reduces the latent heat to a marked degree but it remains considerably higher than the pure alcohols.

The use of methyl or ethyl alcohol, either pure or in solution, presents some additional problems such as inflammability. Methyl alcohol is somewhat toxic; ethyl alcohol is essentially non-toxic.

Liquid oxygen is another fluid that could be used as an expendable coolant. In as much as oxygen is needed for crew breathing during flight, any cooling obtained with the oxygen used for such purposes would be an efficient cooling means. To secure cooling by means of liquid oxygen, the liquid would have to be stored in very well insulated containers at a very low temperature. The critical temperature of oxygen is -182°F. Unless the oxygen is carried at temperatures below the critical temperature, the only cooling that could be obtained would be that available from sensible heating up to the temperature at which it is to be used for crew breathing. The use of oxygen as an expendable coolant in quantities greater than required for purposes such as crew breathing does not appear warranted because the cooling available is less than for a number of other fluids not requiring storage at the very low temperatures. If water, for example, were pre-cooled and stored in an insulated container so as to be initially frozen, approximately one third more cooling could be obtained per pound of weight. The additional cooling is obtained by virtue of the latent heat of fusion and the sensible heat required to increase the temperature of the solid and liquid before vaporization. Pre-cooling of the liquid (or freezing) would enable the expendable coolant system to cool equipment for a limited time under conditions at which the ambient pressure is too high to permit boiling of the liquid.

A simple expendable coolant system is shown schematically in figure 90. In this system, the expendable coolant is conducted to the equipment component through a line by means of pressure or a small pump. Vaporization takes place at the equipment. Each evaporator is equipped with a pressure-regulating valve controlled by the evaporator temperature. This type of system, because of the long lines from stored expendable coolant to the equipment, would require a non-freezing expendable coolant.

An expendable coolant system utilizing a heat transport fluid is shown schematically in figure 91. In this system, the expendable coolant is evaporated in the storage tank. The vaporization is achieved by means of a heat transport fluid flowing through tubes immersed in the expendable coolant. While this system involves a heat transport fluid with lines, pump, etc., it permits the utilization of water as the expendable

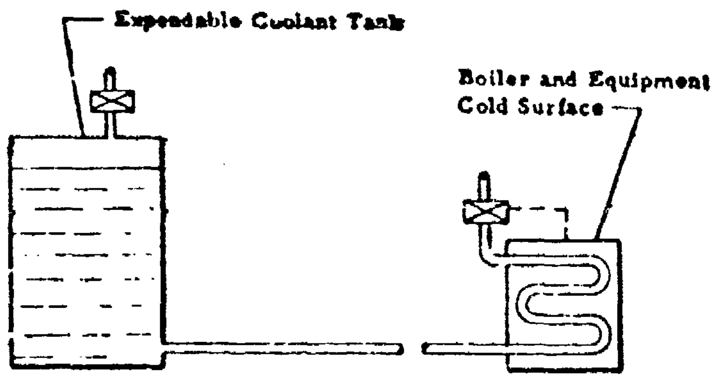


FIGURE 90 WATER-ALCOHOL EXPENDABLE COOLANT SYSTEM

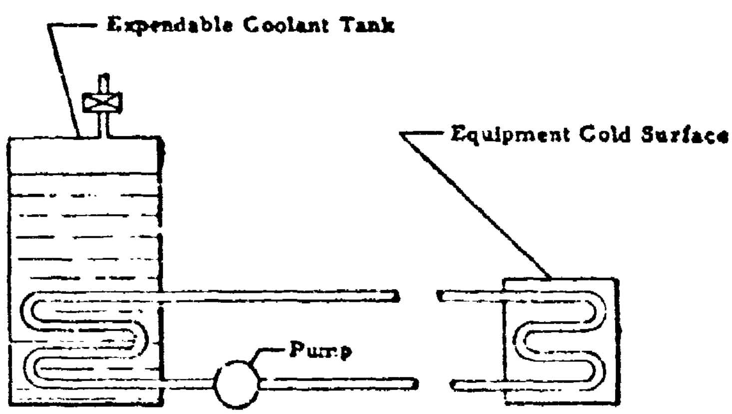


FIGURE 91 WATER EXPENDABLE COOLANT SYSTEM

coolant provided the storage tank is designed so that it will not be damaged by freezing. With this system, the need for pressure control valves at each of several evaporators is eliminated.

B. Results of the Analysis of Expendable Coolant Equipment Cooling Systems

The initial weights of the simple expendable systems depend on the design flight duration. Initial weights of systems for a 10 kw cooling load are plotted versus flight duration in figure 92 for the water system assuming the associated lines, storage tank, etc., weight 20 pounds plus 20% of the fluid weight. The storage tank for this system must be designed so that the water can expand when freezing. The initial weights of a water-ethyl alcohol system are also plotted in figure 92 assuming the lines, pressure regulators, storage tank, etc., weigh 20 pounds plus 15% of the fluid weight.

The weight of expendable coolant cooling systems varies during flight; consequently, for evaluation purposes, some average weight is more indicative of the effect on the aircraft. The evaluation average weight of the two types of simple expendable coolant systems are illustrated in figure 93. The curves were drawn with the same assumptions as the initial weight curves, taking a value of 70% of the required expendable coolant weight as the evaluation average weight. The term "evaluation average weight" is used for all systems utilizing an expendable coolant so as to differentiate systems exhibiting varying weight and having an inherent time limitation from those having a constant equivalent weight independent of time of operation. The evaluation average weight as used in this report includes the actual weight of components, an equivalent weight for power or drag (as discussed in section III), and one-half the weight of the expendable coolant required for the specified flight duration.

C. Conclusions With Regard to Expendable Coolant Cooling Systems

1) Expendable coolant systems are applicable only to flight of relatively short duration.

2) The systems are very simple and involve a minimum of mechanical parts and controls.

3) Water is by far the most efficient coolant on a weight basis.

4) Ammonia may be desirable if very low temperatures are required.

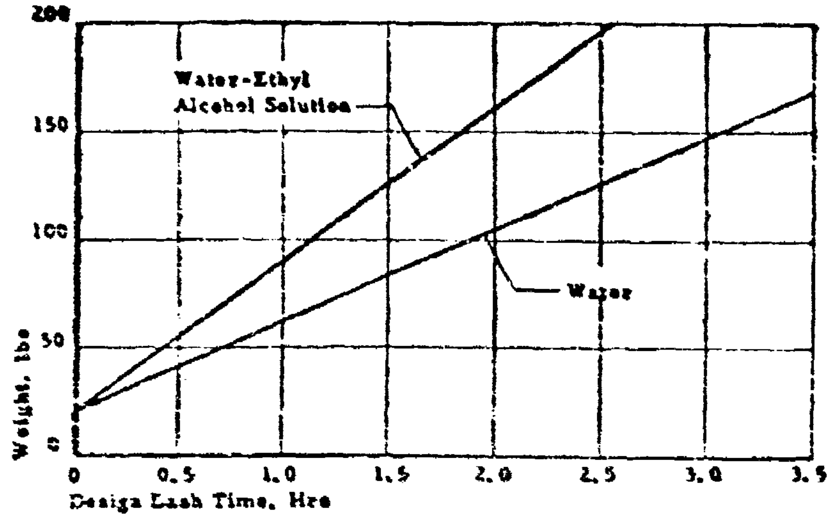


FIGURE 92 INITIAL WEIGHT OF EXPENDABLE EVAPORATIVE COOLANT SYSTEMS

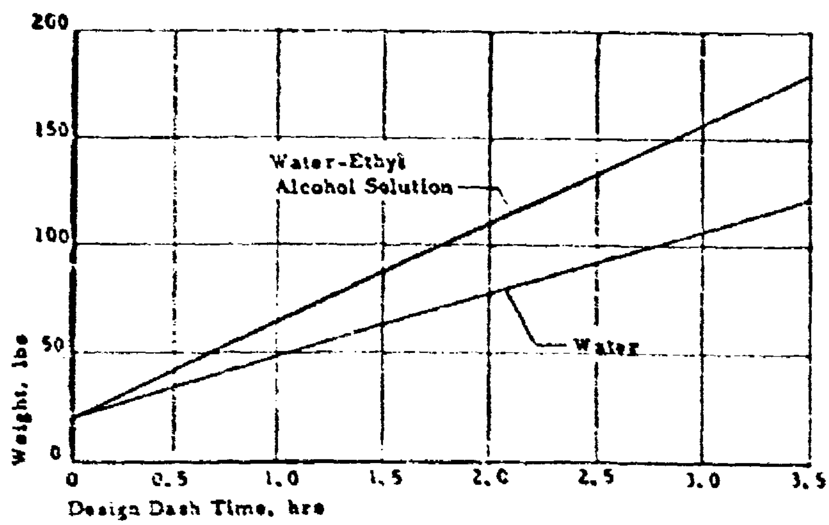


FIGURE 93 EVALUATION AVERAGE WEIGHT OF EXPENDABLE EVAPORATIVE COOLANT SYSTEMS

5) The systems are relatively insensitive to flight or equipment conditions within the attainable temperature range.

6) Ground cooling can be secured provided the equipment temperature level is above the boiling point of the liquid at the ambient pressure.

7) Perhaps the most promising field for expendable coolant cooling systems is in conjunction with another system. Such applications are discussed in the following section of this report.

SECTION VIII

COMBINATION SYSTEMS

A. General Considerations

For some aircraft and flight plans, it may be possible to reduce the total penalty ascribed to the cooling system by utilizing more than one type of cooling system. Two basic conditions that require a combination cooling system, or for which less penalty can be secured by utilizing two or more systems as a composite cooling system, are:

1) systems that possess inherent limitation in temperature difference or in sink temperatures, thus precluding operation for a portion of the altitude and Mach number envelope, and

2) systems that exhibit excessive penalty for a portion of the altitude-Mach number range, or for certain flight patterns.

An example of the first condition would be vapor cycle systems utilizing a refrigerant that has a critical temperature below the maximum sink temperature encountered in the altitude and Mach number of interest.

The second condition can be illustrated by air cycle systems that could be designed for any condition but that may impose an excessive penalty at the higher altitudes and Mach numbers. Simple expendable coolant systems would require excessive weight for long-duration flights and would therefore be included in the second category.

Aircraft equipment cooling systems can be divided into two basic types closely related to the flight design condition:

1) Systems designed for continuous operation

2) Systems designed for a limited operating time

The usual air cycle and vapor cycle systems are examples of systems designed for continuous operation. Any expendable coolant system has an inherent time limitation which is proportional to the amount of coolant, and therefore it must be designed for a limited operating time.

The design of composite systems is dependent on the actual flight velocity, altitude, and duration at the various conditions. In general, any combination system should be designed to utilize each elementary system in an efficient operating range, that is, in flight conditions for which the system imposes a relatively small penalty. The use of an auxiliary system such as an expendable coolant system to extend the range of applicability of a simple ram cooling system is a particular case of a combination system which could be used for a cruise-dash-cruise flight in which the major "cruise" portion of flight would be at a subsonic or a low supersonic velocity with a short duration "dash" at higher velocity. The ram temperature rise, at the high velocity, would preclude the use of the simple system for that portion of the flight. On the other hand, excessive weight would be required for continuous cooling by means of the expendable coolant system.

The analysis in sections V to VII gives an indication of the flight conditions for which combination systems may be necessary or desirable and also indicates the particular systems which would be feasible for such combination systems.

Variations of the basic air cycle and vapor cycle cooling systems are not considered as combination systems in this report.

A combination of an air cycle and a vapor cycle system with an air cycle turbine driving a vapor cycle compressor was considered. This particular system was dropped from further consideration when preliminary calculations indicated relatively little merit and a difficulty in matching the vapor cycle compressor power requirements with the power available from the air cycle turbine at varying conditions. It is believed that the recovery of thrust by means of a centrifugal air compressor is a more practical means of utilizing the air turbine power. Further, the cooling of a vapor cycle condenser by means of cooled air from an air cycle turbine did not appear practical because of the low pressures at the turbine outlet and the heat transfer involved. The condenser must remove the basic equipment cooling load plus the heat added by the compressor. (This latter heat quantity, because of efficiency considerations, will be somewhat greater than the amount of heat extracted by the turbine.)

Many different types of combination systems have been considered. The analysis indicates varying degrees of merit for the various systems. Final judgment as to the worth of a particular system will of course depend on a specific application.

B. Ram Air Cooling System Combined with an Expendable Coolant System

1. Basic Considerations

For aircraft operating on a cruise-dash-cruise flight schedule, a combination ram air-expendable coolant system using air as the cooling medium may be worthwhile for some applications. A system of this type is shown schematically in figure 94. Operation of the system is as follows: Under cruise conditions, ram air enters the intake duct, flows through the equipment component wherein heat generated by the equipment is absorbed, and is then expelled through a convergent nozzle. For dash conditions, the exit duct is closed off (as shown in the figure) and cooling is accomplished by forced circulation of air through a closed circuit, including the equipment component and a boiler heat exchanger, in which heat is removed from the air by boiling off an expendable coolant. The arrangement of the system provides ram pressurization of the flow passages which promotes effective heat transfer and reduces pumping power requirements; also, ram air entering the system to make up leakage losses is cooled in the boiler heat exchanger before entering the equipment component.

In order to secure efficient performance from a system of this type, several important requirements must be met, including:

1) The equipment items comprising the equipment component must be designed to permit effective heat transfer to the cooling air at low pressure drop. In the case of electronic equipment, this goal can best be achieved through the use of a "modular" construction in order to adequately control the airflow over individual equipment items. In general, high heat transfer effectiveness can be achieved by provision of fins for increasing the effective equipment surface area for heat transfer and by arranging the equipment so that the cooling air flows over items having progressively higher temperature tolerances in the direction of flow and leaves the equipment component at the highest possible temperature compatible with effective cooling of items having the highest allowable operating temperatures. Pressure losses can be minimized by passing the cooling air through a number of equipment items (or modules) in parallel. From the standpoint of cooling effectiveness, much is to be gained by locating equipment items requiring cooling in a compact package, in so far as this is possible.

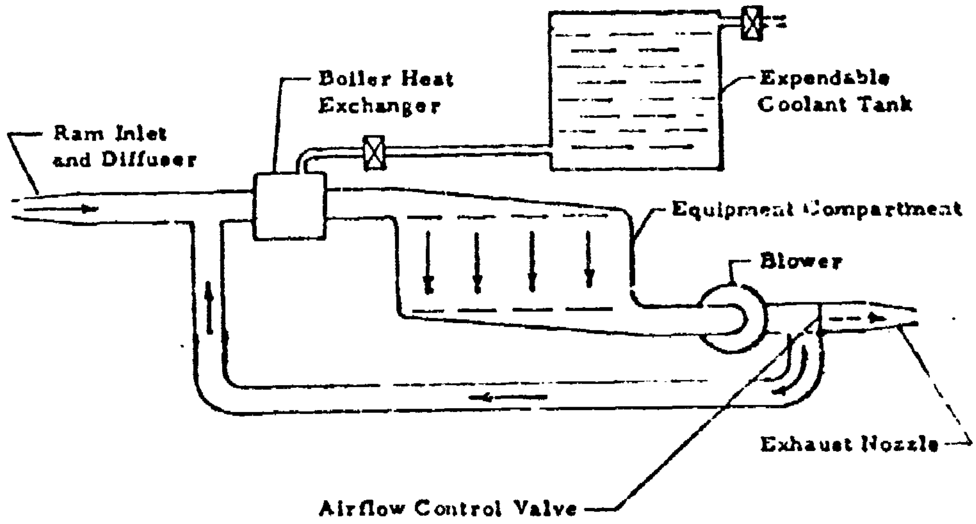


FIGURE 94 COMBINATION RAM AIR AND EXPENDABLE COOLANT SYSTEM

2) Provision of an efficient ram air intake in the proximity of the equipment component. In some instances (e.g., for small high performance aircraft) cooling air might be obtained from the engine inlet duct. Generally, however, an individual inlet would be necessary for each equipment package.

3) For dash conditions, the cooling air should enter the equipment component at the lowest practicable temperature; otherwise, prohibitively high pumping power requirements may result because of the large air circulation rate required. Therefore, the expendable coolant used with a system of this type must have a fairly low boiling point. Fortunately, the boiling point decreases with increasing altitude, resulting in a reduction of the required airflow rate which compensates in part for the increased pressure losses at high altitude caused by the reduced air density. The relative merits and effect of various properties of a number of expendable fluids are discussed in the portion of this report on expendable coolant cooling systems (section VIII). For systems using air as a heat transport fluid, the expendable coolant must have a relatively lower boiling point than for a system utilizing a liquid heat transport fluid. This is because the weight flow rate times specific heat (wc_p) is usually much less for air than for liquids; consequently, air will have a greater temperature change and therefore a lower equipment inlet temperature (T_{Ei}).

In as much as the overall performance of the cooling system depends to a great extent upon the properties of the expendable coolant used, careful consideration must be given to the selection of that coolant. It should be remembered as pointed out above that this particular system imposes more stringent requirements as to boiling temperature than do other expendable coolant systems considered in this study.

In view of the above considerations and the general discussion in section VII, the alcohols are considered to be the most satisfactory expendable fluids for use in the ram air expendable coolant systems for aircraft use whereas ammonia may be preferable for certain missile applications and for the lower equipment temperatures.

2. Analysis of Dash Conditions

Under high speed flight conditions, cooling is achieved by forced circulation of air through a closed circuit including the equipment and a heat exchanger, in which a suitable expendable coolant is boiled off in order to remove heat from the circulating air (see figure 94). The initial step in this analysis is to determine the required air circulation rate, w_a.

Equating the equipment heat rejection rate to the rate of heat absorption by the circulating air

$$w_a = \frac{3.95\,kw}{T_{Ee} - T_{Ei}}$$

The equipment inlet temperature T_{Ei} can be expressed in terms of conditions existing in the boiler heat exchanger. Defining the boiler heat exchanger effectiveness as

$$e_{bo} = \frac{T_{bo-i} - T_{Ei}}{T_{bo-i} - T_{bo}} \qquad (140)$$

The equipment inlet air temperature is

$$T_{Ei} = e_{bo}\,T_{bo} + (1 - e_{bo})\,T_{bo-i} \qquad (141)$$

where T_{bo} is the boiling temperature of the expendable coolant and T_{bo-i} is the temperature of the air entering the boiler heat exchanger. Assuming that ram air enters the system at a rate zw_a due to leakage and as the blower and motor also increase the temperature of the air,

$$T_{bo-i} = (1 - z)\,T_{Ee} + z\,T_T + \frac{2.95\,HP_b}{\eta_M\,w_a} \qquad (142)$$

By making use of the above equations, the expression for the required airflow rate can be written in this form:

$$w_a = \frac{3.95\,kw + 2.95\,(1 - e_{bo})\,HP_B/\eta_M}{T_{Ee} - e_{bo}\,T_{bo} - (1 - e_{bo})\,(1 - z)\,T_{Ee} + z\,T_T} \qquad (143)$$

The required expendable coolant is

$$W_{exp} = \frac{3600\,Du\,(wc_p)_a}{L}\,(T_{bo-i} - T_{Ei}) \qquad (144)$$

or $$W_{exp} = \frac{3600\ Du}{L}\left[0.95kw\left(1 + \frac{z(T_T - T_{Ec})}{T_{Ec} - T_{Ei}}\right) + \frac{0.708HP_B}{\eta_M}\right] \qquad (145)$$

Referring to appendix I, the weight of an aluminum, extended-surface boiler heat exchanger is given by the expression

$$W_{bc} = 1.42\ w_a\ \Omega_{bo}^{1.317}\ \phi_{bo}^{-0.317} \qquad (146)$$

where Ω_{bo} is a function of the heat exchanger effectiveness e_{bo} and ϕ_{bo} is a pressure loss parameter which is related to the pressure drop of air flowing through the exchanger

$$\frac{\Delta p}{p_i} = \frac{\bar{\theta}_{bo}}{(\delta_{bo-i})^2}\ \phi_{bo} \qquad (147)$$

Equation (146) is plotted versus ϕ_{bo} is figure 63 for several values of exchanger effectiveness. The boiler heat exchanger is illustrated schematically in figure 60. Ram air entering the system to make up leakage losses causes a momentum drag:

$$D_{mom} = 34.7\sqrt{\theta_a}\ Mw_a \qquad (148)$$

The blower pressure rise factor $(\Delta p/p_i)_B$ is related to the pressure losses in the boiler heat exchanger and the equipment component (neglecting the minor ducting losses) by this equation:

$$\left(\frac{\Delta p}{p_i}\right)_B = 1 - \frac{\left[1 - \left(\frac{\Delta p}{p_i}\right)_{bo}\right]\left[1 - \left(\frac{\Delta p}{p_i}\right)_E\right]}{\left[1 - \left(\frac{\Delta p}{p_i}\right)_{bo}\right]\left[1 - \left(\frac{\Delta p}{p_i}\right)_E\right]} \qquad (149)$$

Values of $(\Delta p/p_i)_{bo}$ and $(\Delta p/p_i)_E$ appearing in this equation can be determined in either of two ways (1) these pressure loss ratios can be treated as design variables and assigned arbitrary values independent of altitude and Mach number or (2) they can be assigned specific values for a particular design altitude and Mach number. In the latter case, the pressure drop and heat transfer effectiveness for both the heat exchanger and the equipment component must be determined in terms of airflow rate, pressure level and mean temperature for "off-design" conditions. This approach is of course more representative of the performance of an actual system which must operate over a wide range of flight conditions with fixed geometry. The analysis of off-design conditions is discussed below.

Using data from reference 1, the required shaft power and weight for an efficient centrifugal blower operating at approximately 60% of maximum capacity are

$$
hp_B = 87.4\, \theta_{Bi}\, w_a\, (\Delta p/p_i)_B^{-0.6} \quad (150)
$$

and

$$
W_B = 0.313\, \frac{w_a^{1.2}\, \theta_{Bi}^{.6}}{\left(\delta_{Bi}\right)^{1.2}} \left(\frac{\Delta p}{p_i}\right)_B^{-0.6} \quad (151)
$$

where θ_{Bi} denotes blower inlet conditions. As shown in section V, the weight of an a-c motor plus the additional generator weight required to supply power to the motor can be expressed as

$$
W_{BM} = 5 + 3.1\, hp_B \quad (152)
$$

In evaluating the performance penalty imposed on the aircraft by the cooling system, the effects of momentum drag and required electrical power are expressed in terms of an equivalent weight as described in section III. The total effective weight of the cooling system, for evaluation of the aircraft performance penalty, is found by adding the actual or equivalent weights for each item. The evaluation average weight for the expendable coolant is equal to the mean expendable coolant weight, $W_{exp}/2$ plus the weights of storage container, piping, etc. Assuming the latter amounts to 20% of the initial coolant weight, the net weight which can be attributed to the expendable coolant is

$$
\overline{W}_{exp} = 0.7\, W_{exp} \quad (153)
$$

where W_{exp} is found from equation (145). In addition, the weight of ducting required for ram cooling during cruising conditions must be included. This is discussed subsequently under the cruising condition analysis.

For high speed flight conditions, the equipment cooling system is pressurized by the ram pressure rise at the diffuser inlet (see figure 94). Assuming a normal shock entry, the inlet total pressure p_T° can be expressed by the relationship (reference 23):

$$
p_T = p_a \left(1 + \eta_{diff} \frac{1 + 0.2M^2}{7M^2 - 1}\right)^{3.5} \left[1 + 1.167(M^2 - 1)\right] \quad (154)
$$

For ... is plotted in figure 64 for an assumed diffuser efficiency ... ram pressure rise based upon the above equation would correspond to a reasonably efficient leading edge or nose inlet. Since ram pressurization becomes very important to the operation of the cooling system at high altitudes, an inlet with poor pressure recovery would severely limit the performance of the system.

For off-design conditions, the effectiveness e_{bo} and the pressure loss factors $(\Delta p/p_i)$ and $(\Delta p/p_i)_E$ must be computed before the analysis described above can be carried out. Generally, the heat transfer effectiveness for heat exchangers involving a single flowing fluid can be expressed in terms of the fluid flow rate, w, as follows:

$$1 - \exp\left[-\Omega'\left(\frac{w'}{w}\right)^j\right] \qquad (155)$$

where the primed quantities are considered to be design values. The exponent j depends upon the characteristics of the heat exchanger core; for a given core, j is a known (positive) constant. The parameter Ω' is related to the design effectiveness e' through the relationship

$$\Omega' = -\ln(1 - e') \qquad (156)$$

From equation (155) it is apparent that the effectiveness decreases with increasing flow rate and vice versa. Strictly speaking, the preceding equations are valid only when heat transfer occurs from the flowing fluid to an isothermal surface. This condition is closely approached in the boiler heat exchanger because of the relatively high boiling heat transfer coefficient on the water side. The pressure drop factor for heat exchangers can usually be expressed with sufficient accuracy in the form

$$\frac{\Delta p}{p_i} = \left(\frac{\Delta p}{p_i}\right)' \frac{\theta}{\theta'}\left(\frac{\delta_i'}{\delta_i}\right)^2\left(\frac{w}{w'}\right)^j \qquad (157)$$

provided that turbulent flow is maintained and the pressure drop remains small relative to the initial pressure.

The analysis of off-design flight conditions can be carried out with sufficient accuracy for the present purpose by using equations (155) to (157). The required ... rate w_a is first formed from equations (153), (155), and (156). ... en $(\Delta p/p_i)_{bo}$ and $(\Delta p/p_i)_E$ are found ... design values by applying equation (157). In practice, changes in e_p can be ignored over the ... flow rates encountered with the ... non-expendable coolant system. However, the pressure loss factor ... widely with changes in altitude and Mach number, with an appreciable effect upon the overall system performance. Results of off-design ... tions are discussed

[illegible]

[illegible] described previously [illegible] the cruising [illegible] For choked [illegible] flow rate is represented [illegible] by the [illegible]

$$[illegible] \qquad (159)$$

[illegible] dependence of the flow [illegible] (see [illegible] since the [illegible] nozzle, [illegible] rate given by equation (92) is obtained for [illegible] cruising condi[illegible]

[illegible]

[illegible] Mach number [illegible]

$$[illegible] \qquad (166)$$

[illegible]

[illegible] temperature, Mach number and altitude, and [illegible]

[illegible]

[illegible] speed [illegible] cooling [illegible]

Cooling load

$$kw = 10 \text{ at } T_{E_e} = 275\,°F$$

Dash (M = 1.6 at 60,000 feet)

$$\left(\frac{\Delta p}{p_i}\right)'_E = 0.05 \text{ and } w_a' = 0.266 \text{ lb/sec}$$

$$\epsilon_{bo} = 0.9 \text{ at } (\Delta p/p_i)_{bo} = 0.05 \text{ and } w_a' = 0.266 \text{ lb/sec}$$

$$\left(\frac{\Delta p}{p_i}\right)_E = 0.0328 \quad \frac{w_a^{1.8}}{(\delta_T)^{2.0}} \quad \text{equation with } j = 1.8$$

Cruise (M = 1.4 at 60,000 feet)

$$w_a = 0.40 \text{ lb/sec}$$

$$\left(\frac{\Delta p}{p_T}\right)'_{dt} = 0.04 \qquad \ell_{dt} = 12 \text{ ft}$$

In addition to the above data, a total leakage area of 0.075 in^2 was assumed, corresponding to z = 0.1 for sea level flight at M = 1.2. Also, the weight of the blower motor and the increased generator weight required to supply power to the blower (equation 152) were determined for the conditions M = 1.8 at 70,000 feet. This enables the cooling system to operate at 70,000 feet for M = 1.8 or greater.

The [illegible] component pressure drop of $(\Delta p/p_i)_E = 0.05$ at $w_a = 0.266$ for the design condition (M = 1.6 at 60,000 feet) is equivalent to a pressure drop of 2 inches of water for the same flow rate under standard sea level conditions. The flow rate of $w_a = 0.266$ corresponds to the required flow rate, determined by using equation (145), for a cooling system using ethyl alcohol as the expendable coolant. The effects of changes in $(\Delta p/p_i)_E$ are discussed subsequently.

Characteristics and effective weight penalty of the ram air-expendable coolant system with ethanol coolant. The effective weight penalty for an ethyl alcohol system was determined as a function of Mach number and altitude for the above conditions, using the method of analysis presented above. The results are shown in figure 95 for Du = 1 hr and in figure 96 for Du = 1/2 hr.

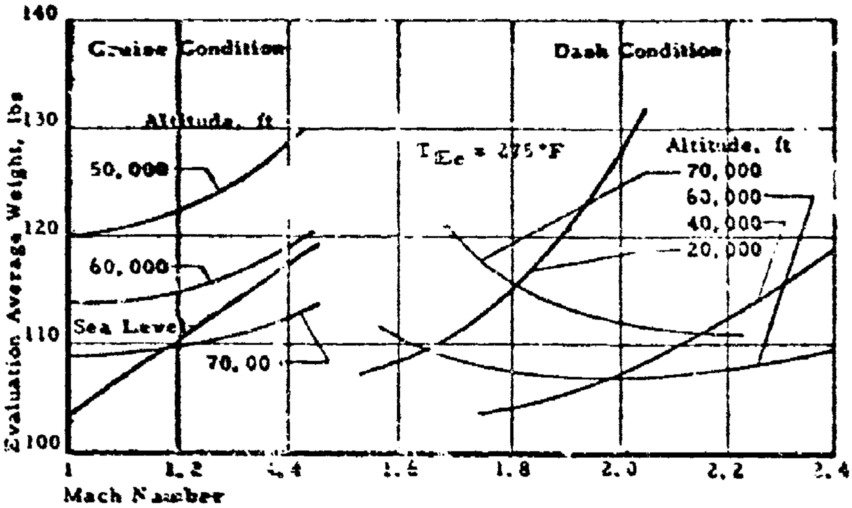


FIGURE 95 EVALUATION AVERAGE WEIGHT OF A RAM AIR COOLING SYSTEM COMBINED WITH AN ETHYL ALCOHOL EXPENDABLE COOLANT SYSTEM (DASH DURATION 1 hr)

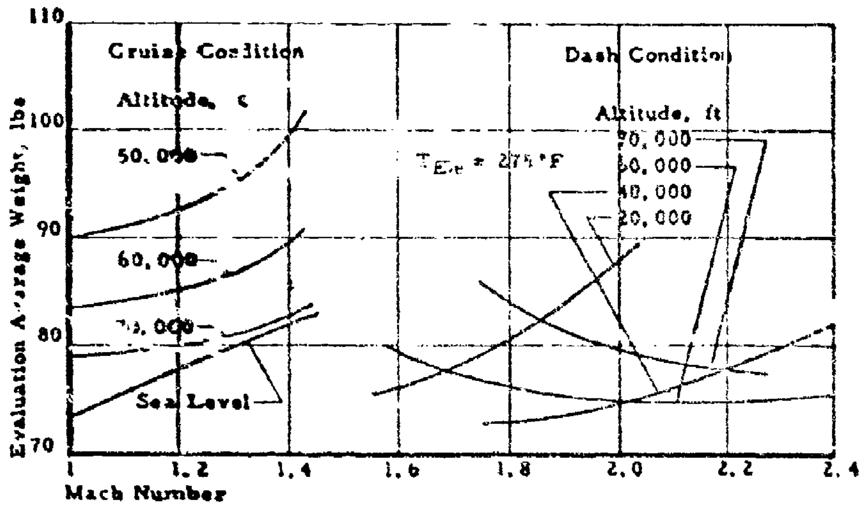


FIGURE 96 EVALUATION AVERAGE WEIGHT OF A RAM AIR COOLING SYSTEM COMBINED WITH AN ETHYL ALCOHOL EXPENDABLE COOLANT SYSTEM (DASH DURATION 1/2 hr)

For cruising condi[illegible]ions, it was assumed that the effective weight due to the expendable c[illegible]olant could be based upon the mean expendable weight given by equati[illegible] (153).

The rapid increas[illegible] in effective weight penalty with increasing Mach number at low a[illegible]titudes is the result of ram air infiltration caused by leakage whi[illegible] imposes both a momentum drag and an increased cooling load u[illegible]on the cooling system. This trend is reversed at high altitudes becau[illegible] of the reduced ram pressurization at low speeds which increases [illegible] the blower power requirement (figures 97 and 98).

The relative merits [illegible] of methyl alcohol and ethyl alcohol expendable coolants from the effecti[illegible]e weight standpoint have been examined for the conditions listed abov[illegible]. The comparative cooling system weight penalties for these coolant[illegible] are shown in figures 99 and 100 for flight at an altitude of 60,000 fee[illegible] and D_u = 1/2 hr. The variation in required blower power with altitude a[illegible]d Mach number for the coolants is shown in figures 97 and 98. The low[illegible]er power requirement for methyl alcohol results from the lower boiling [illegible]mperature of this fluid, permitting reduced flow rates and lower pr[illegible]ssure losses. The effects of changing $(\Delta p/p_i)_E$ from the value $(\Delta p/p_i)_E$ [illegible] 0.05 given above are shown in figure 100 for the dash condition M [illegible] 1.6 at 60,000 feet. Other things being equal, the system effective weig[illegible] penalty is essentially proportional to the cooling load. The weight pe[illegible]alty decreases with increased allowable equipment component exit temp[illegible]rature, i.e., with increasing temperature tolerance for the hottest equip[illegible]ent items being cooled.

Because of the relatively small variation [illegible] cooling system effective weight penalty with changes in altitude and Mach [illegible] number (for the range of conditions considered herein) as shown in figu[illegible]s 95 and 96 for the ethyl alcohol system, the results discussed above [illegible] considered fairly representative of the individual effects of the c[illegible]oling system variables over the full range of flight conditions shown in [illegible]ure 3.

5. Conclusions With Regard to the Ram Air Cooling [illegible] Combined With an Expendable Coolant System

From the foregoing results, the following observations [illegible] made with regard to the performance and possible application of the ram air [illegible] expendable coolant cooling system:

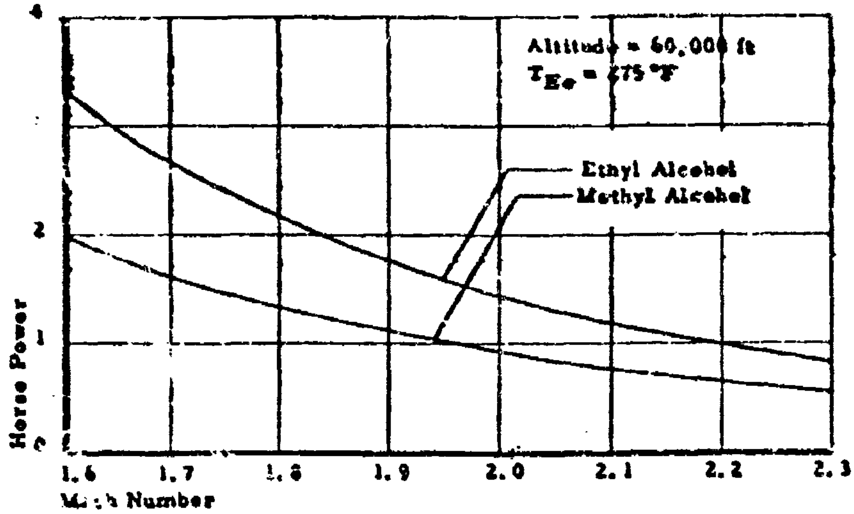


FIGURE 97 BLOWER POWER VERSUS MACH NUMBER FOR A RAM AIR COOLING SYSTEM COMBINED WITH AN EXPENDABLE COOLANT SYSTEM

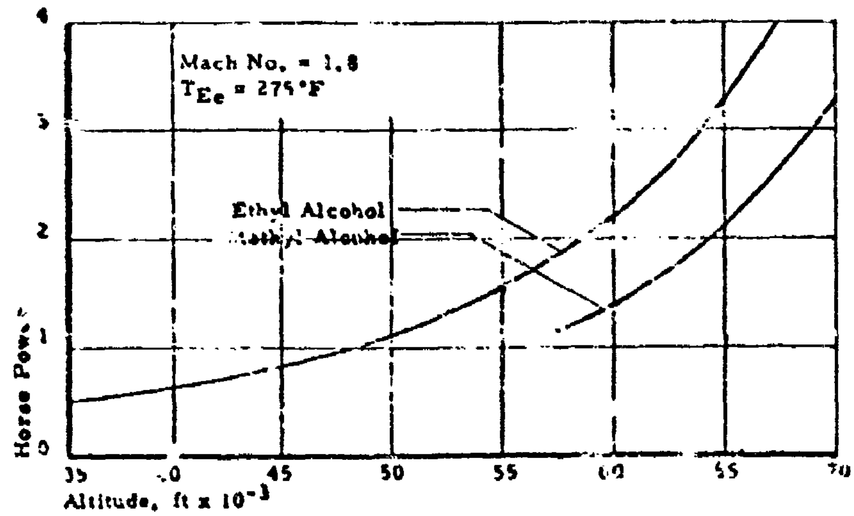


FIGURE 98 BLOWER POWER VERSUS ALTITUDE FOR A RAM AIR COOLING SYSTEM COMBINED WITH AN EXPENDABLE COOLANT SYSTEM

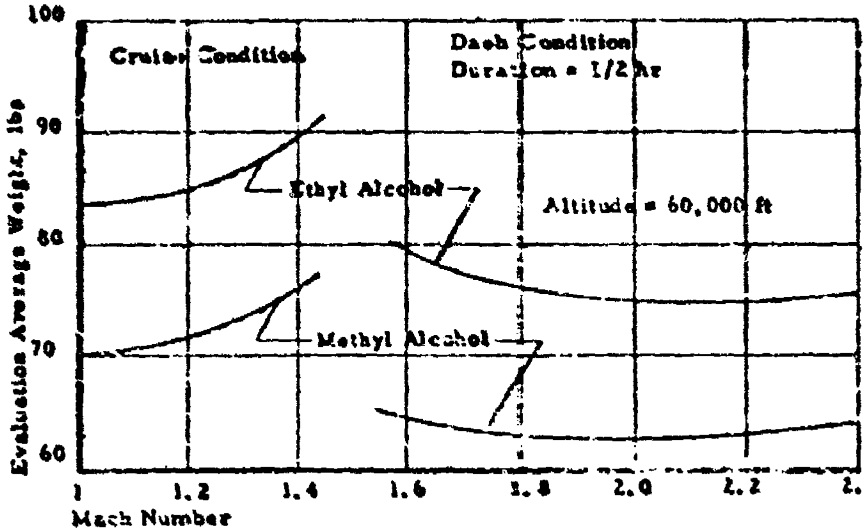


FIGURE 99 EVALUATION AVERAGE WEIGHT VERSUS MACH NUMBER OF A RAM AIR COOLING SYSTEM COMBINED WITH AN EXPENDABLE COOLANT SYSTEM

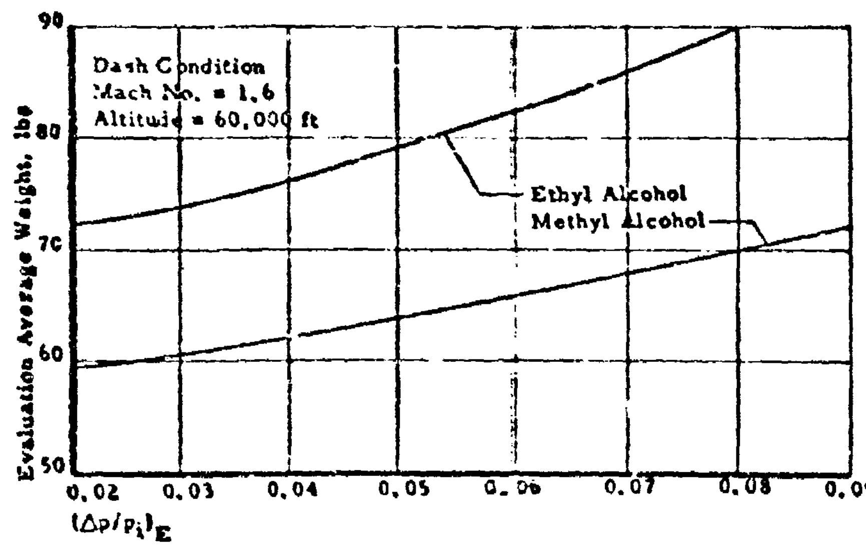


FIGURE 100 EVALUATION AVERAGE WEIGHT VERSUS EQUIPMENT PRESSURE DROP OF A RAM AIR COOLING SYSTEM COMBINED WITH AN EXPENDABLE COOLANT SYSTEM

1) The system provides cooling at relatively low penalty for both dash and cruise conditions for a total dash duration on the order of one hour or less. The most advantageous application would be for flight schedules involving long cruise duration at speeds up to M = 1.4 (above 40,000 feet) coupled with comparatively short dash duration.

2) Both methyl alcohol and ethyl alcohol appear to be satisfactory as expendable coolants with the former having an appreciable superiority because of its higher latent heat of vaporization and lower boiling point.

3) This cooling system is well adapted to individualized applications where several widely separated equipment packages require cooling.

4) Because of the relatively low inlet temperature of the cooling air, a cooling system of this type can effectively cool equipment items having low, as well as medium and high, temperature tolerances.

5) The air-cooled package could be designed to permit easy access; also, the cooling system is not highly vulnerable to battle damage. Ram air constitutes a very dependable source of cooling for cruise conditions.

6) In the final analysis, applicability of the system depends upon design provisions for effective cooling of heat-generating equipment by means of airflow with moderate pressure drop, together with proper distribution of the air to the individual items requiring cooling. Small auxiliary blowers could be employed where necessary to facilitate proper airflow distribution in the equipment component.

7) It is quite important that the equipment component (and ducting for the closed circulating system) be well sealed against leakage to prevent ram air infiltration at high speeds.

C. Simple Air Cycle Cooling System Combined With an Expendable Coolant System

The simple air cycle analyzed in section VI of this report can be designed to provide relatively efficient cooling for flight at Mach numbers up to 1.6 except at low altitudes or altitudes above 60,000 feet. For flight above M = 1.6 or H = 60,000 feet, the total equivalent weight imposed by the simple system becomes excessive. A means of reducing the total equivalent weight for those cases where the flight time at the dash condition is relatively short is provided by adding a boiler heat exchanger and an expendable coolant.

1. Basic Considerations

A simple air cycle cooling system combined with an expendable coolant system is illustrated schematically in figure 71. The following analysis would likely require modification before applying it to a different configuration. A system of this type can utilize water as the expendable coolant because the boiling takes place before the turbine inlet and, consequently, is at a high temperature. The system would have to be designed to prevent damage should freezing occur and to assure a self-thawing unit, e.g., by having the water container located so as to be thawed out by bleed air if frozen.

The amount of air that must be bled from the turbine compressor is, in terms of the equipment exit temperature and inlet temperature,

$$w_b = \frac{3.95kw}{T_{Ee} - T_{Ei}} \tag{92}$$

In this case, the equipment inlet temperature is again equal to the turbine exit temperature but the turbine inlet temperature is reduced by the heat of vaporization of the expendable coolant that is boiled off. That reduction in temperature is

$$T = \frac{L\, W_{exp}}{865\, t\, w_b} \tag{161}$$

The required bleed air rate is then

$$w_b = \frac{3.95kw}{T_{Ee} - \left[(T_{ti})_{W_{exp} = 0} - \frac{L\, W_{exp}}{865\, t\, w_b} \right] \lambda t} \tag{162}$$

where $\lambda = 1 - \eta_t (1 - r^{-0.286})$

or, in terms of the bleed rate without an expendable coolant,

$$w_b = (w_b)_{W_{exp} = 0} - \frac{L\, W_{exp}}{\left[T_{Ee} - (T_{ti})_{W_{exp} = 0} \right] 865\, t} \tag{163}$$

The temperature of the turbine exit (and equipment inlet) will not be permitted to drop below 32°F. This stipulation is made to preclude the possibility of an icing problem at the turbine exit and in ducts from the turbine to the equipment. With this assumption, just enough water will be boiled off to secure a turbine inlet temperature such that, with the available pressure ratio, the turbine exit temperature is at 32°F.

It should be noted that with the above assumption, since the equipment exit temperature is taken as an independent variable of analysis, the bleed rate will be constant as indicated by equation (92). The amount of expendable coolant that must be provided will depend on the dash duration and the reduction of temperature necessary.

$$W_{exp} = 865\, t \left[\frac{3.95kw - w_b \left[T_{Ee} - (T_{ti})_{W_{exp}=0} \right]}{L\, \lambda} \right] \quad (164)$$

2. Results of the Analysis

The evaluation of the simple air cycle combined with an expendable coolant system is similar to the analysis of the simple air cycle systems in so far as the air cycle portion is concerned except for the reduced air bleed. In addition to the weight and air bleed of the air cycle system, the combination system imposes an additional weight by virtue of the boiler heat exchanger, the tank and lines required for the expendable coolant and the coolant that must be carried aboard the aircraft.

These systems are evaluated on the basis of an evaluation average weight defined as the sum of the total equivalent weight of the air cycle portion, the weight of the boiler heat exchanger, and 0.6 of the weight of expendable coolant required for the dash portion of the flight.

$$W_{ev-a} = W_{T-eq} + W_{bo} + 0.6 W_{exp} \quad (165)$$

The systems have been evaluated on the basis of a dash time of one hour for a cooling load of 10 kw. Because of the assumption that the turbine exit temperature is 32°F, the weight of expendable coolant as given by equation (164) will be zero for any condition for which the pressure ratio and heat exchanger characteristics are such that the turbine exit temperature for the simple system is 32°F. For such conditions, a simple air cycle cooling system of the assumed design is therefore adequate for continuous cruise cooling. A system that will provide cooling for one hour at specified conditions will provide continuous cooling for somewhat less stringent conditions. It is assumed in all calculations that water is to be used as the expendable coolant because of its high latent heat of vaporization.

The evaluation average weights of the simple air cycle combined with an expendable coolant system such as illustrated in figure 71 have been determined for a broad range of conditions. The assumed operating conditions are equipment exit temperature, T_{E_e} = 275°F, cooling load, kw = 10, dash duration, t = 1 hr. Conditions are therefore the same as for the simple air cycle discussed previously except for the limited flight time which applies to any system utilizing an expendable coolant.

The total evaluation weights are plotted versus Mach number in figures 101 and 102 for drag equivalent weight factors of two and three respectively. The amount of expendable coolant that must be boiled off for various conditions is indicated in figure 103. The variation of evaluation average weight with altitude is shown in figure 104. Figures 105 and 106 indicate the operating limits for simple air cycle cooling systems combined with an expendable coolant system. The expendable system in each case is designed for a one-hour dash at the specified altitude and Mach number. In this case, as the expendable coolant is used to cool bleed air, the dash flight conditions are an important factor as indicated by figures 101 to 106. In this connect, the simple air cycle combined with an expendable coolant is different from other expendable coolant systems in which the expendable coolant is used more directly to cool equipment and the amount of coolant required is therefore essentially independent of flight conditions.

3. Conclusions With Regard to Simple Air Cycle Cooling Systems Combined With Expendable Coolant Systems

1) The combined system maintains the general characteristics of the simple air cycle. For dash times of approximately one hour, the variation of the evaluation average weight with Mach number and with altitude is similar but reduced by a very significant amount for conditions for which the simple air cycle imposes a high total equivalent weight.

2) The smaller equivalent weights are primarily due to a reduction in bleed and ram airflow requirements.

3) The combination system is a simple means of extending the short time, high speed dash capabilities of the simple air cycle system.

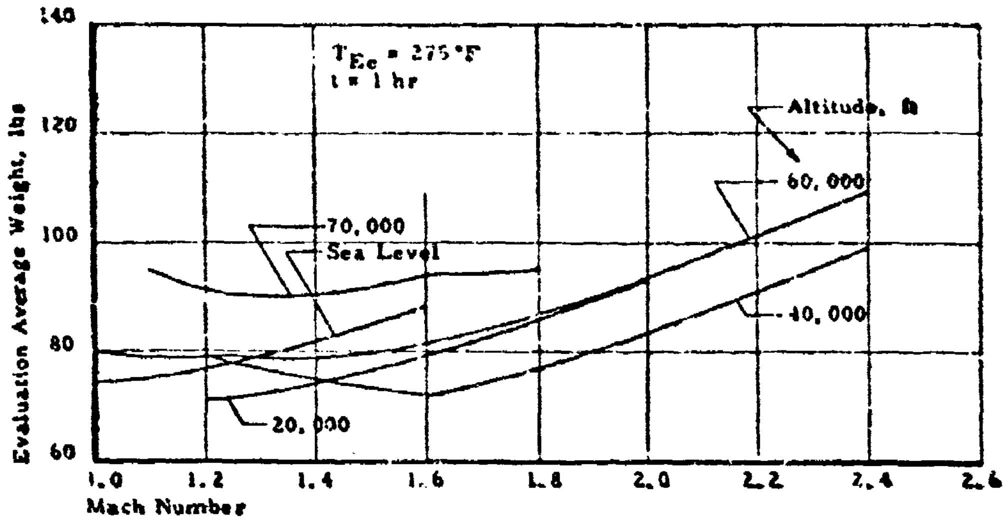


FIGURE 101 EVALUATION AVERAGE WEIGHT OF A SIMPLE AIR CYCLE COOLING SYSTEM COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (f=2)

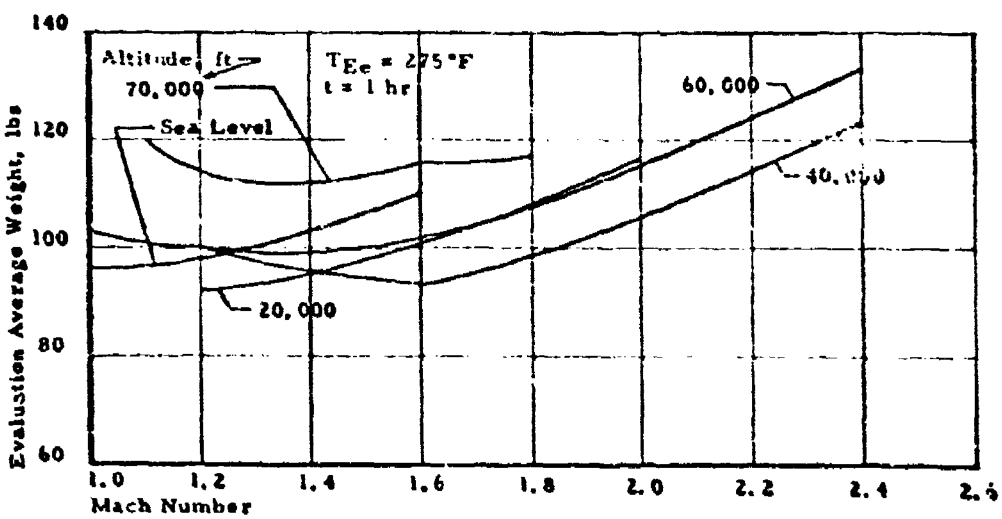


FIGURE 102 EVALUATION AVERAGE WEIGHT OF A SIMPLE AIR CYCLE COOLING SYSTEM COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (f=3)

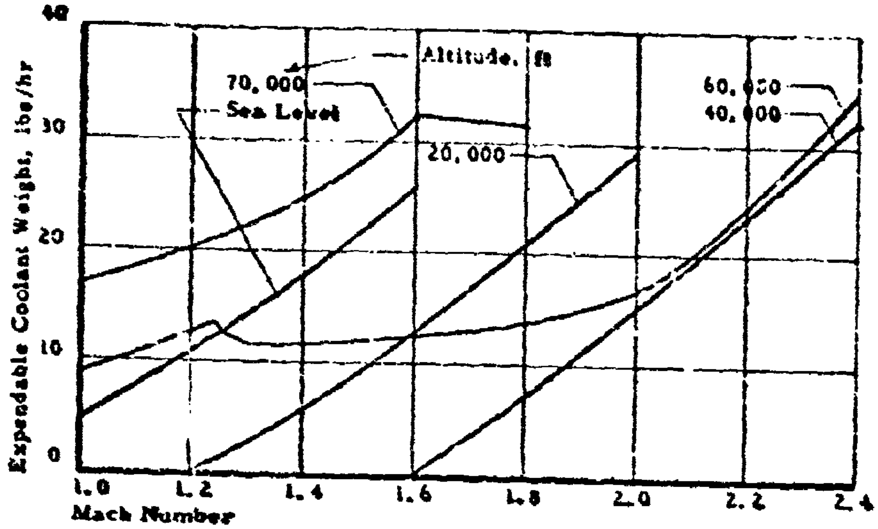


FIGURE 103 WATER REQUIREMENTS FOR A SIMPLE AIR CYCLE COOLING SYSTEM COMBINED WITH AN EXPENDABLE COOLANT SYSTEM

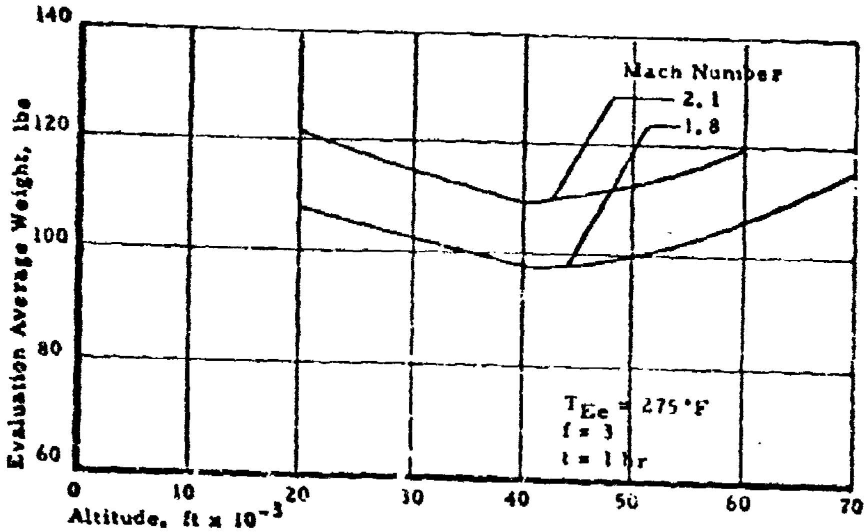


FIGURE 104 EVALUATION AVERAGE WEIGHT OF A SIMPLE AIR CYCLE COOLING SYSTEM COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM

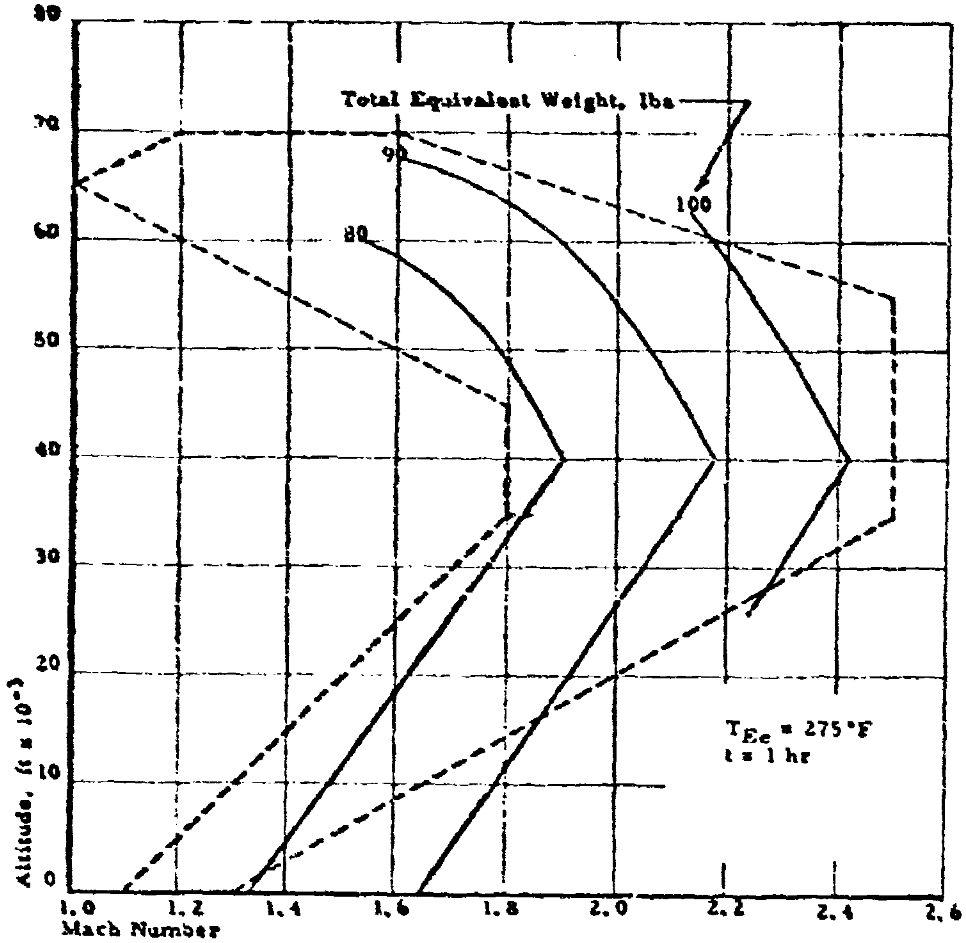


FIGURE 105 COOLING SYSTEM OPERATING LIMITS FOR A GIVEN TOTAL EQUIVALENT WEIGHT FOR A SIMPLE AIR CYCLE COOLING SYSTEM COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (f = 2)

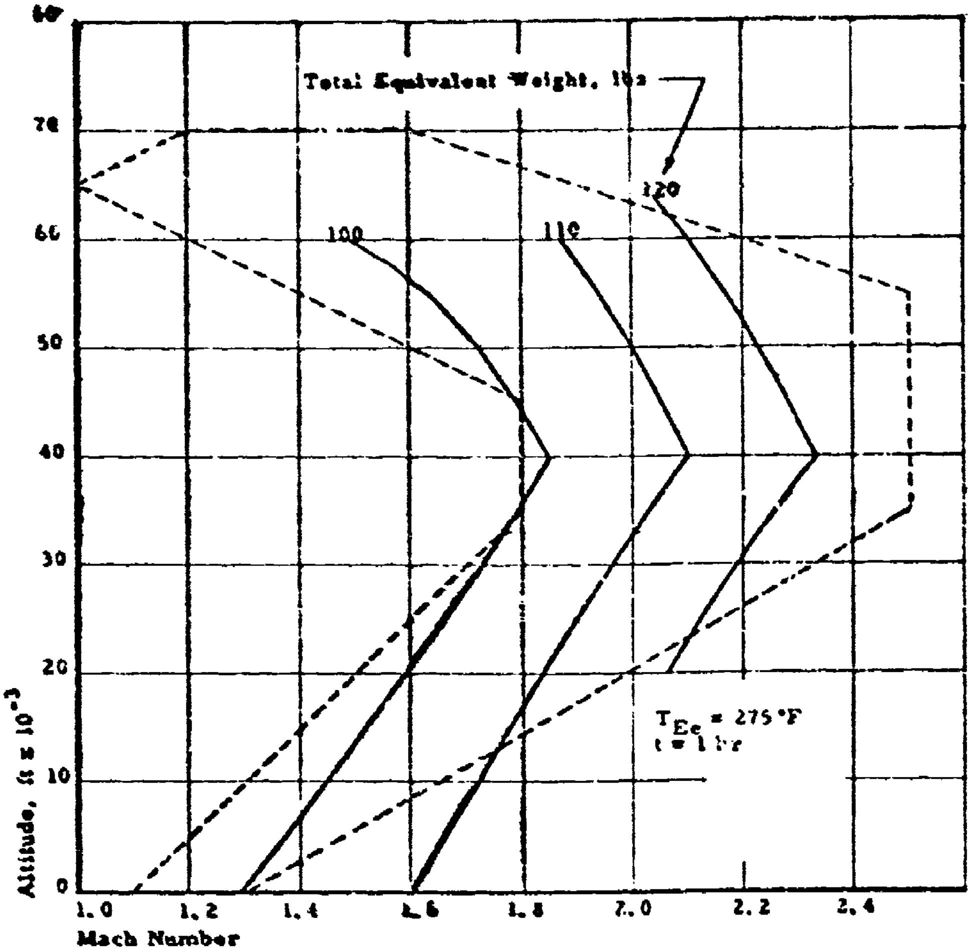


FIGURE 106 COOLING SYSTEM OPERATING LIMITS FOR A GIVEN TOTAL EQUIVALENT WEIGHT FOR A SIMPLE AIR CYCLE COOLING SYSTEM COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (f = 3)

D. Surface Heat Exchanger Combined With An Expendable Coolant Cooling System

One of the combination cooling systems analyzed in this study is comprised of a heat transport fluid system with a surface heat exchanger and an expendable coolant system. Such a system is applicable for cruise flight conditions up to Mach 1.5 and for any dash condition of limited duration.

1. General Considerations

The surface heat exchanger has been analyzed in a manner entirely analogous to that used for the surface condenser and heat exchangers as described in section V of this report. Essentially the same assumptions have been made.

The weight of the surface heat exchanger is primarily dependent on the area and on the material and type of construction. As with a condenser, the heat transfer coefficient on the inside of the passages is very large compared with the external heat transfer coefficient. Therefore the resistance to heat flow is essentially determined by the external heat transfer coefficient.

A detailed analysis of this type of heat exchanger is included in section V on vapor cycle surface condensers. The weight of the surface heat exchanger is

$$W_{ex-S} = \frac{F_S \, C \, kw}{h_o (T_{Ee} - T_r)} \tag{166}$$

where $F_S = \dfrac{h_o (s_d + s)}{U (s_d + s)} \dfrac{(T_{Ee} - T_r)}{(T_{Ee} - T_{Ei})} \ln\left(\dfrac{T_{Ee} - T_r}{T_{Ei} - T_E}\right)$

The constant C in equation (166) includes the weight per square foot and a factor to convert kw into Btu/hr. Note that in this case the heat exchanger inlet temperature is equal to the equipment exit temperature and the heat exchanger exit temperature is equal to equipment inlet temperature, hence the terms T_{Ee} and T_{Ei}. If, as is often the case for relatively high fluid flow rates, the fluid temperature change $(T_{Ee} - T_{Ei})$ is small in comparison with the difference between the recovery temperature and the fluid exit temperature $(T_{Ee} - T_r)$, then assuming that U is approximately equal to h_o and that the fin effectiveness is approximately one, the factor F_S will be only slightly greater than one. The weight of a surface heat exchanger can then be approximated by the equation

$$W_{ex\text{-}S} \doteq \frac{C\ kw}{h_o\,(T_{Ee} - T_r)} \qquad (167)$$

Equation (167) may be used for initial calculations and as an indication of effects of the various factors.

The surface heat exchanger portion of this cooling system can provide cooling for any flight velocities for which the recovery temperature is sufficiently below the exit temperature of the equipment, the difference required is dependent on area and on the heat transfer coefficient. The surface area required is, with the same assumptions used in equation (167),

$$A_{ex\text{-}S} \doteq \frac{Q}{h_o\,(T_{Ee} - T_r)} \qquad (168)$$

The required area for surface heat exchangers is given versus Mach number for various altitudes in figure 112.

Cooling of equipment can be achieved with this type of cooling system without excessive areas for cruise up to Mach 1.6 at the higher equipment temperatures. The dash flight must, of course, be limited in duration, the time depending on the expendable coolant used and on the initial weight.

2. Results of the Analysis

A schematic diagram of the system using a water-ethyl alcohol solution as the expendable coolant and as the heat transport fluid is illustrated schematically in figure 107. The system, modified for the use of water as the expendable coolant, is shown in figure 108. The weights of surface heat exchanger and expendable coolant combination systems designed for a 10 kw cooling load are plotted versus dash time for various conditions in figures 109 and 110 for water and in figure 111 for a water-ethyl alcohol solution. The approximate required surface area is plotted versus Mach number for equipment exit temperatures of 160°, 210°, and 275°F in figure 112. The cruise conditions for which the various systems will provide continuous cooling are specified on the figures.

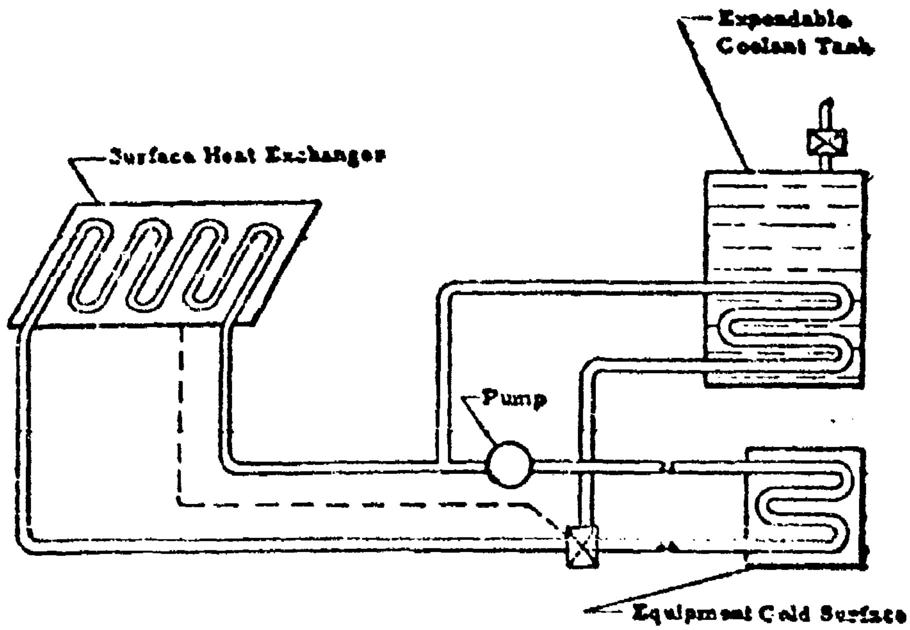


FIGURE 107 SURFACE HEAT EXCHANGER COMBINED WITH WATER EXPENDABLE COOLANT SYSTEM

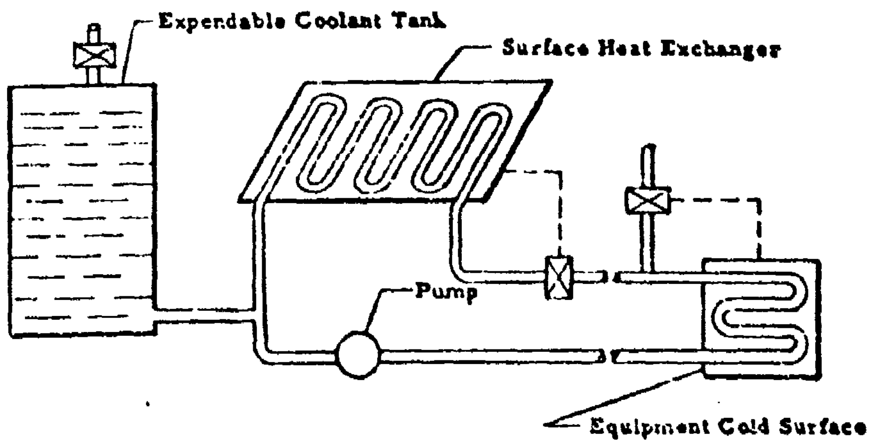


FIGURE 108 SURFACE HEAT EXCHANGER COMBINED WITH WATER-ALCOHOL EXPENDABLE COOLANT SYSTEM

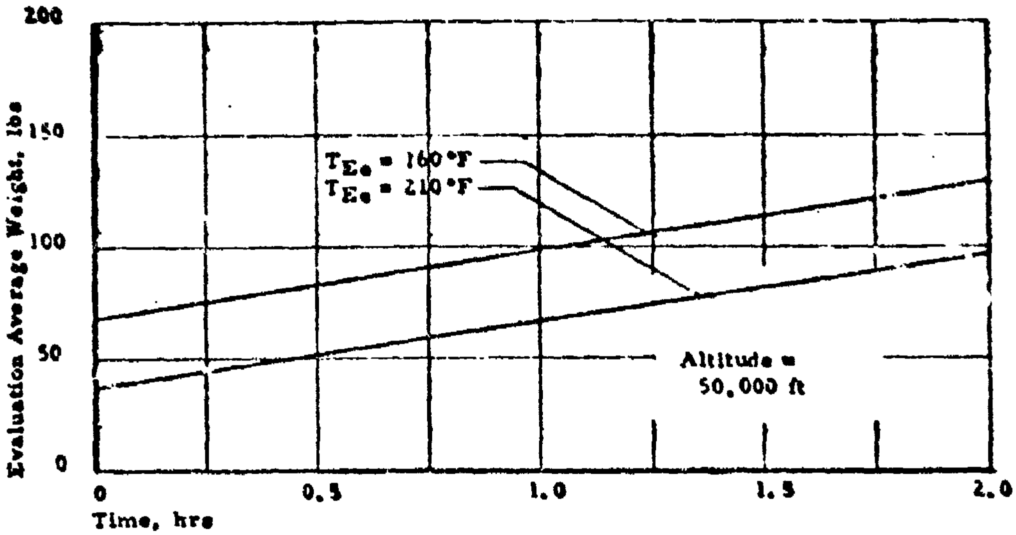


FIGURE 109 EVALUATION AVERAGE WEIGHT OF A SURFACE HEAT EXCHANGER COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 1.5)

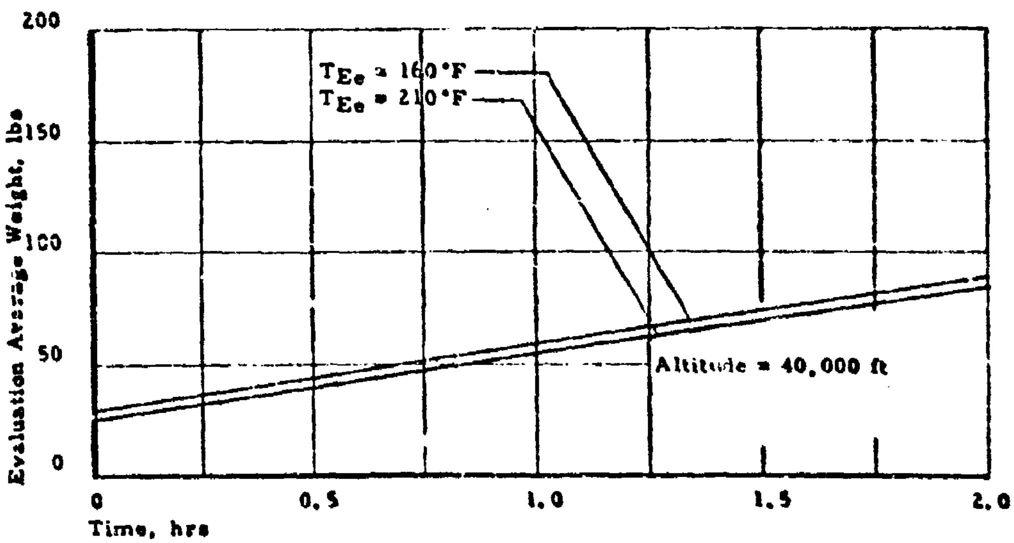


FIGURE 110 EVALUATION AVERAGE WEIGHT OF A SURFACE HEAT EXCHANGER COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 1)

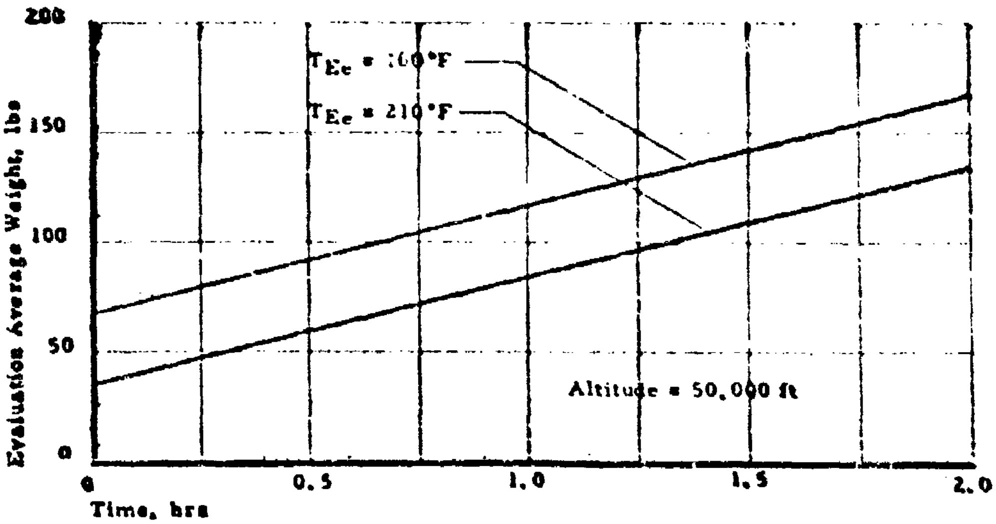


FIGURE 111 EVALUATION AVERAGE WEIGHT OF A SURFACE HEAT EXCHANGER COMBINED WITH A WATER-ETHYL ALCOHOL EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 1.5)

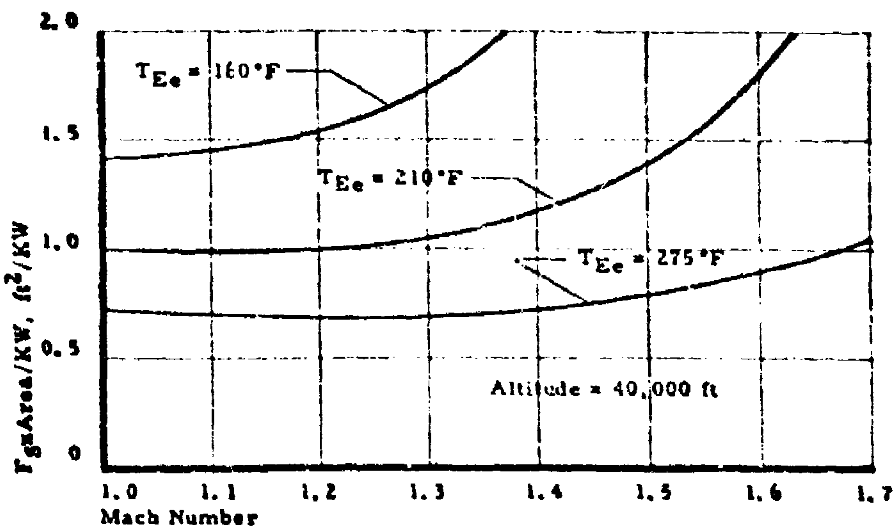


FIGURE 112 SURFACE HEAT EXCHANGER AREAS VERSUS MACH NUMBER

3. Conclusions With Regard to Surface Heat Exchanger Expendable Coolant Systems

The following conclusions can be drawn from the preceding analysis:

1) The surface heat exchanger combined with an expendable coolant provides an efficient means for cooling aircraft equipment provided that the required surface area is available.

2) The evaluation average weights are under 100 pounds for dash times of up to one hour. The weights do not include the heat transfer surface at the equipment.

3) The system weight is not greatly affected by the dash flight conditions unless heat is transferred to the stored fluid and lines from the surroundings.

4) The system imposes a small power requirement and surface drag (as pointed out in section V) during cruise operation. During high speed dash, the surface exchanger is not used; consequently, no drag is imposed under the flight conditions for which drag is most significant.

E. Vapor Cycle Cooling Systems Combined With Expendable Coolant Systems

1. Basic Considerations

Vapor cycle cooling systems were considered in detail in section V of this report. It was noted that vapor cycle systems have an inherent limitation on the condenser temperature depending on the critical temperature of the refrigerant and on the chemical stability of the refrigerant at the higher temperatures. In the case of Freon-11, the limit is below the maximum considered in this study. The use of an expendable coolant for the high speed dash portion of a flight is a possible means of extending the range of a vapor cycle cooling system that uses a refrigerant with a critical temperature below the maximum temperatures that are necessary for the heat pump cycle. Expendable coolants can also be used to extend the temperature limits of a system that is not adequate for other reasons, e.g., a system in which the compressor is not designed to compress to the pressure required under certain conditions.

The expendable coolant portion can be utilized in two different ways:

1) To cool the condenser of a vapor cycle system at any time when the ram temperature (or the recovery temperature) is above the system operating limits.

2) To cool the heat transport fluid directly and thus actually operate as a complete system in place of the vapor cycle system for a temperature beyond the capabilities of the vapor cycle system.

The first method imposes a greater demand on the expendable coolant system because the heat added by the vapor cycle compressor and its motor must be absorbed in addition to the actual equipment cooling load. This concept also requires operation of the vapor cycle system at all times and therefore involves the vapor cycle power requirement during high speed dash flight.

The second concept actually involves two separate cooling systems in so far as operating characteristics are concerned. The vapor cycle cooling system for cruise flight is separate from an expendable coolant system for the limited-duration high speed dash. With this method, the expendable system is actually a substitute system and is only required to absorb the equipment cooling load. The vapor cycle system would be shut off and therefore not require power nor impose a surface (or ram) drag during the dash portion of the flight. This method does, however, require an expendable coolant that will boil at a temperature below the allowable equipment temperature because the heat is not pumped to a higher temperature level as is the case with the condenser cooling concept. This latter requirement is not considered serious since the high speed dash would likely be at high altitude where the boiling point even of water, the preferred coolant, would be low enough to be used.

Because of the above considerations, systems in which the expendable coolant is used to cool the heat transport fluid directly will in general impose a smaller equivalent weight. Consequently, in this study, that is the configuration assumed. A system utilizing a water-alcohol expendable coolant is illustrated schematically in figure 113. A system in which the heat transport fluid is conducted to a water tank so that water can be used even under sub-freezing conditions is illustrated in figure 114.

2. Results of Analysis

The evaluation average weight of vapor cycle cooling systems combined with expendable coolant systems depends on the duration of the dash portion of the flight, the expendable coolant used, the cruise design conditions and the design details of the various components. This particular type of system is simply a combination of two of the simple systems previously analyzed and actually involves very little modification of either system.

The evaluation average weights of 10 kw Freon-11 vapor cycle systems designed for cruise cooling at 50,000 feet altitude at Mach 2.0 and at Mach 1.6 combined with an expendable coolant system for higher speed dash are plotted versus dash time in figures 115 and 116. The evaluation average weights for water cycle systems designed for cruise at M = 1.8 and M = 2.2 combined with an expendable coolant system for any higher speed dash are plotted versus dash time in figures 117 and 118. Curves for equipment temperatures of 160°, 210°, and 275°F are included.

The evaluation average weights given by figures 115 to 118 apply to flight at cruise conditions, one half before the expendable coolant is used and one half after the expendable coolant is used. The evaluation average weight applicable for the high speed dash portion of the flight would be reduced approximately 5 or 10% because power would not be required to drive the vapor cycle compressor at that time. The fact that any surface drag imposed by a surface condenser has been neglected in the vapor cycle analysis would not be a factor for the high speed dash because the condenser is not dissipating the heat while the expendable coolant is being used.

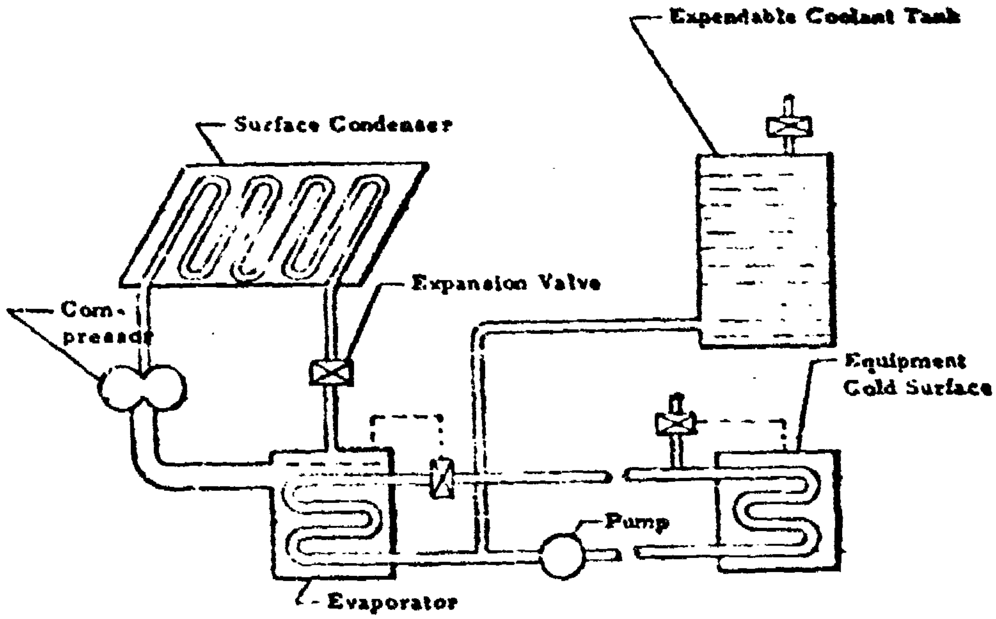


FIGURE 113 VAPOR CYCLE COOLING SYSTEM COMBINED WITH WATER-ALCOHOL EXPENDABLE COOLANT SYSTEM

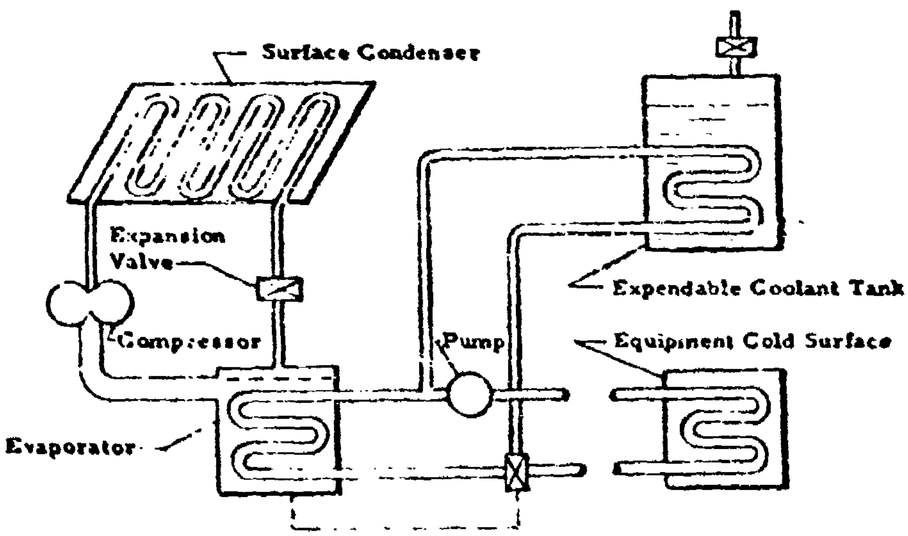


FIGURE 114 VAPOR CYCLE COOLING SYSTEM COMBINED WITH WATER EXPENDABLE COOLANT SYSTEM

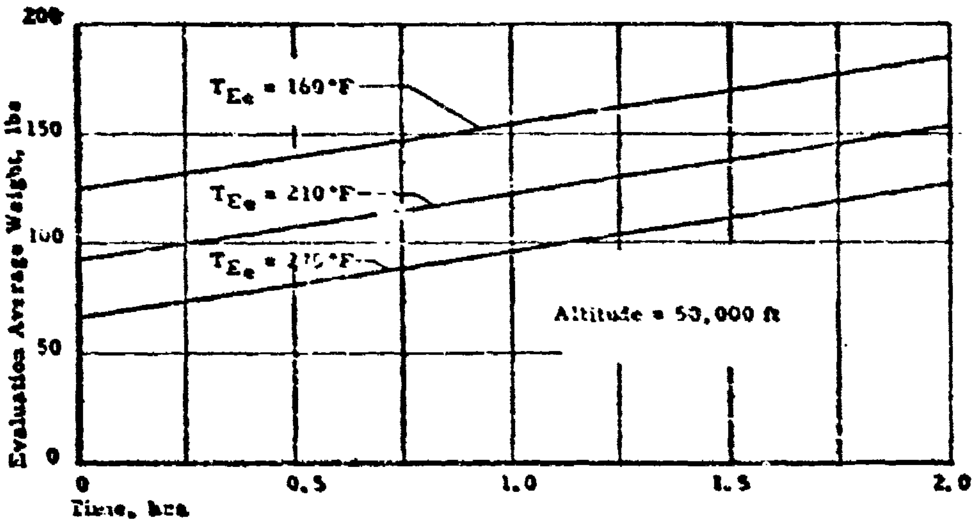


FIGURE 115 EVALUATION AVERAGE WEIGHT OF FREON-11 VAPOR CYCLE COOLING SYSTEMS COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 2.0)

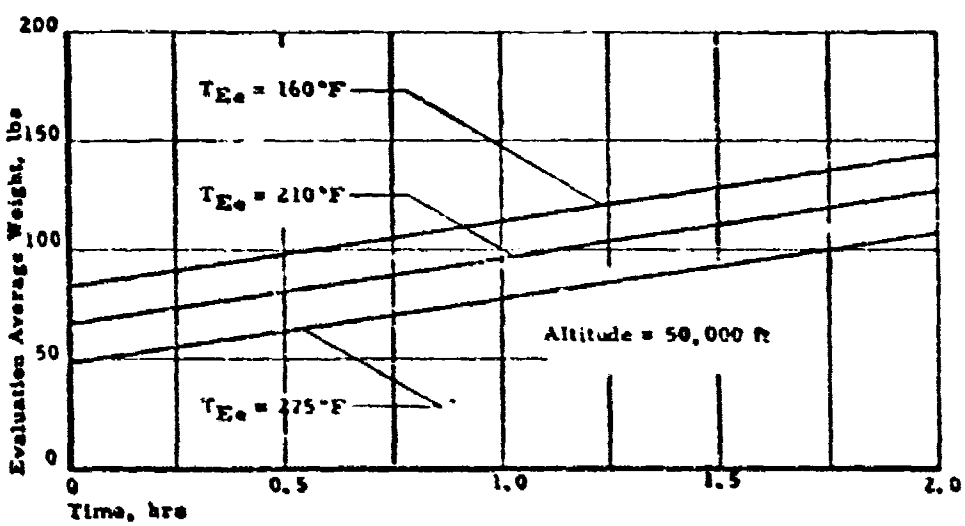


FIGURE 116 EVALUATION AVERAGE WEIGHT OF FREON-11 VAPOR CYCLE COOLING SYSTEM COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 1.6)

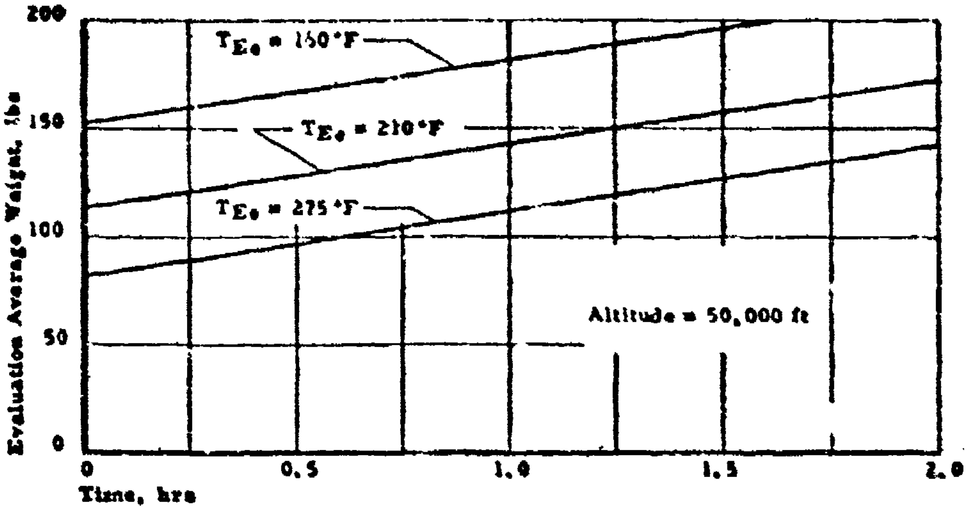


FIGURE 117 EVALUATION AVERAGE WEIGHT OF WATER VAPOR CYCLE COOLING SYSTEMS COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 2.2)

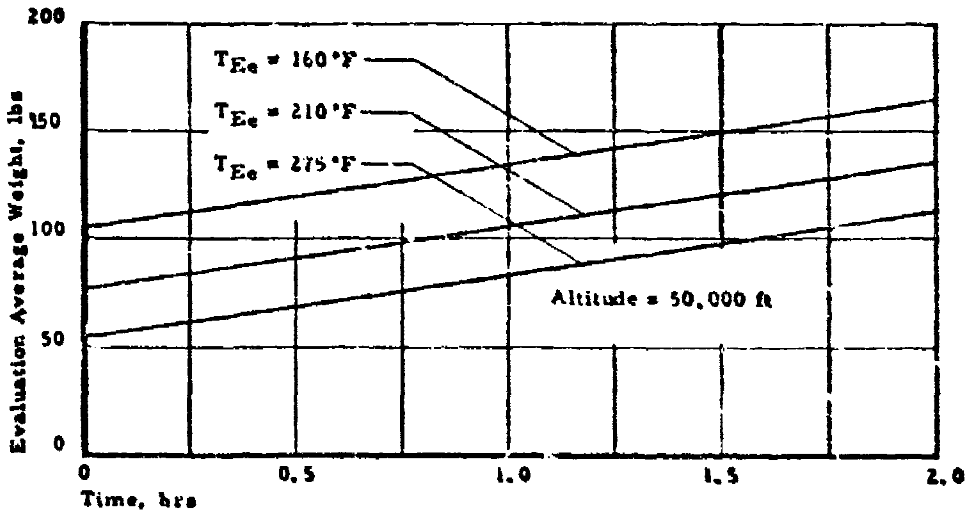


FIGURE 118 EVALUATION AVERAGE WEIGHT OF WATER VAPOR CYCLE COOLING SYSTEMS COMBINED WITH A WATER EXPENDABLE COOLANT SYSTEM (CRUISE MACH NO. = 1.8)

3. Conclusions With Regard to Vapor Cycle Cooling Systems Combined With Expendable Coolant Systems

The following conclusions can be drawn from the analysis of cooling systems comprised of a vapor cycle system and an expendable coolant.

1) The use of an expendable coolant provides a simple means of increasing the maximum temperature capabilities of a vapor cycle cooling system for a limited time of operation.

2) The evaluation average weights depend primarily on the cruise conditions for which the vapor cycle system is designed and on the dash duration for which expendable coolant is provided.

3) During high speed dash flight, the systems impose no drag and only a small power requirement for circulating the heat transport fluid.

4) The water vapor cycle systems would require some engineering development before actual working systems could be built.

F. Other Combination Equipment Cooling Systems

A considerable amount of heat can be absorbed by the engine fuel even with a relatively small increase in the fuel temperature. The engine fuel is therefore a very attractive heat sink; however, the fuel is normally used for other cooling purposes such as oil cooling and usually is heated up to very near the maximum allowable temperature by such heat sources. The use of turbine engine fuel as a heat sink for equipment cooling has not, therefore, been analyzed in detail.

The fuel flow to the afterburner presents a possibility. Such fuel flow cannot be used for a continuous cooling function at relatively low speeds because the afterburner is not normally used for such cruise flight. Fuel to afterburner, therefore, may not be suitable for cooling purposes such as oil cooling. The fuel to the engine afterburner could be used to provide a heat sink for equipment cooling for flight at very high speeds that require the additional thrust attainable through use of an engine afterburner. This is particularly attractive in that such cooling would be available for the portion of a flight schedule which imposes the most difficult conditions for equipment cooling. In this case, it is assumed that cruise cooling could be obtained by means of a very simple system, such as a surface heat exchanger or a ram air system, with the fuel to afterburner being used during high speed dash flight.

Additional cooling would have to be provided for the period when the afterburner is turned off and until flight conditions and temperatures were such that the simple surface or air system would suffice. A system of this general type, utilizing a surface heat exchanger and water as the expendable coolant, is illustrated schematically in figure 119. It is important to note that with this type of system expendable coolant would have to be provided for only a transient condition that would probably not last more than 20 to 30 minutes.

The specific heat of jet fuel is approximately 0.5 Btu/lb °F at 100°F (reference 6). The specific fuel consumption for a typical jet engine afterburner at Mach 2 as given in reference 23 is about 2.5 lb/hr, per pound of thrust. The increase in fuel temperature to the afterburner, assuming all fuel to the afterburner is heated by the cooling load, is

$$\Delta T_f = \frac{3413 \text{ kw}}{SFC_{aft}\ T_{aft}\ c_{pf}} \tag{169}$$

or, taking the values as given above,

$$\Delta T_f = \frac{2730 \text{ kw}}{T_{aft}} \tag{170}$$

Equations (169) and (170) indicate that for a thrust augmentation of about 3000 pounds, the cooling of the equipment by fuel to an afterburner will result in a fuel temperature increase of about 1°F per kilowatt.

The evaluation average weight of a system of this type would be relatively small because of the absence of drag at high velocities and because the amount of expendable coolant required is relatively low.

Many other cooling configurations could be devised using fuel as a heat sink, such as simple fuel heat exchangers, fuel used in combination with air cycle systems, etc. As pointed out above, the use of fuel would of course depend on temperature limitations. The fuel does present a very attractive heat sink, but it perhaps simply is not available because of other considerations.

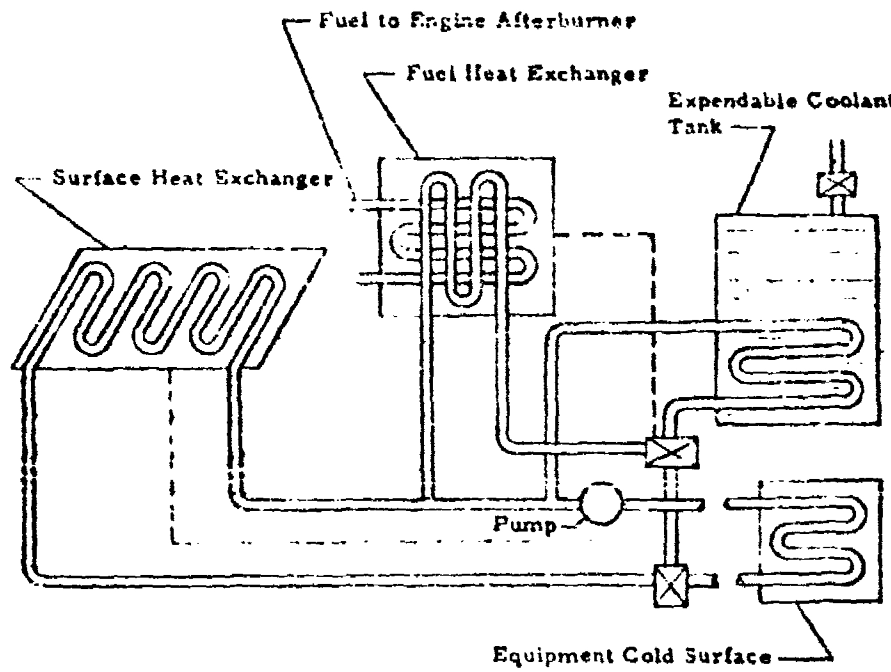


FIGURE 119 SURFACE HEAT EXCHANGER COMBINED WITH FUEL HEAT EXCHANGER AND WATER EXPENDABLE COOLANT SYSTEM

SECTION IX

EVALUATION AND APPLICABILITY OF DIFFERENT TYPES OF EQUIPMENT COOLING SYSTEMS

The determination of the range of applicability and the evaluation of different types of cooling systems depends on a large number of factors and on the relative importance of those factors. In the analysis of the various systems, the equivalent weight or an evaluation average weight has been determined and is presented as a function of velocity and altitude for a number of conditions. In this section, the characteristics of the different systems with respect to those factors will be compared and other typical characteristics of the systems will be pointed out. Among the more significant external factors which affect the design and operation of a cooling system are equipment temperature, flight velocity, and flight altitude. In this section, some of the particular attributes and liabilities of the equipment cooling systems analyzed previously will be compared with regard to these factors, followed by a general comparison of such cooling system characteristics as total equivalent weight requirements, vulnerability, dependability, and applicability to centralized and individualized cooling concepts.

A. Equipment Temperature

It was pointed out in section IV that different types of systems possess very different characteristics with regard to temperature variation of the cooling medium. Simple expendable coolant systems such as illustrated in figure 90 provide a surface that is approximately isothermal. The vapor cycle cooling systems will, in general, have a heat transport fluid temperature variation of 10°F or less. The regenerative air cycle system uses a heat transport fluid but with an equipment inlet temperature 100°F or more below the exit temperature. In this study, the equipment exit temperature has been selected as an independent variable. It should be noted that with a given equipment exit temperature, systems in which the cooling medium has a rather broad temperature variation can, with properly designed equipment cool certain elements to well below the exit temperature. The simple air cycle systems normally cool the equipment by direct circulation of air and therefore do not utilize a heat transport fluid. Such systems will, for many operating conditions, have an air temperature difference between equipment inlet and outlet of over 200°F. If necessary, such systems can cool equipment to very low temperatures. The average temperature of the cooling medium at the equipment is approximately the same for a simple air cycle with an exit temperature at 275°F as for a vapor cycle system with the equipment exit temperature at 160°F. A comparison on this basis must take into account the actual temperature tolerances of equipment elements. The air cycle system, for example, would be capable of cooling about half the cooling load to 160°F or below; the balance of the equipment, located nearer the exit, could be cooled only to from 160° to above the 275°F exit temperature.

In addition to the wide variation in the temperature conditions of the cooling medium at the equipment, the different systems have different characteristics as to the effect of the equipment temperature requirements.

Vapor cycle systems, because of the major effect of evaporator temperature on compressor size and power requirements, are quite sensitive to variations in equipment exit temperature. It is, in fact, this temperature dependence that dictates the small temperature variation at the equipment. The total equivalent weight of a vapor cycle system is approximately double for an equipment exit temperature of 160°F as compared with an exit temperature of 275°F. Systems with a refrigerant with a relatively low latent heat of vaporization have a definite limit to temperature difference between evaporator and condenser (unless a cascade cycle such as the one illustrated in figure 36 is used) which may be a decisive factor. In some applications, this factor may establish a lower limit as to the evaporator temperature for a given condenser temperature.

Simple air cycle systems are also relatively sensitive to equipment temperature tolerances though somewhat less so than vapor cycle systems. The broad variation in the temperature of the cooling medium would likely eliminate the need for equipment exit temperatures as low as might be required for a vapor cycle system.

Regenerative air cycle systems are usually less sensitive to the equipment conditions than are vapor or simple air cycle systems. Figure 88 indicates that an increase in equipment exit temperature from 220° to 340°F results in an equivalent weight decrease of about 20% for a regenerative air cycle system.

Expendable coolant systems or the expendable coolant portion of combination systems using expendable coolants are not greatly affected by equipment conditions provided that the change is within the range of the coolant boiling point. In the event that a different coolant is required, a very marked change in weight may occur. The increase which will accompany a decrease in the equipment temperatures is caused by the reduced cooling effect and by increased heat exchanger weight that may be required.

Ram air type cooling systems or systems using surface heat exchangers are greatly affected by equipment temperature. In fact, the equipment temperature defines the upper limit on flight conditions beyond which cooling is impossible with such systems.

Other effects of equipment temperature are noted in the discussion of the various systems. Specific effects can be determined by re rring to the figures indicating the performance of the various systems.

B. Flight Velocity

The different cooling systems are affected very differently by the aircraft flight velocity. The general characteristics of the systems considered in this study are discussed below. Specific ititative data are included with the analyses of the various systems.

Cooling systems utilizing ram air directly for equipment cooling have a definite limit and the weight will increase very rapidly as the limit is approached. The same is true of simple surface heat exchanger cooling systems.

Simple air cycle cooling systems are relatively unaffected by velocity up to a point depending on the altitude and then the weight increases very rapidly (figures 72 and 73) because the system is dependent on ram air for pre-cooling the compressor bleed air. The simple air cycle cooling system combined with an expendable coolant system is not as greatly affected provided the dash time is not over one hour.

The regenerative air cycle cooling system is affected by flight velocity to a much smaller degree than the simple air cycle as illustrated by figures 83 and 84. The weight of such systems remains quite reasonable even at the higher velocities considered.

The total equivalent weight of Freon-11 vapor cycle cooling systems increases very rapidly with velocity, the limit being well below the maximum velocity being considered in this study. The water vapor cycle system increases, but at a nearly uniform rate, up to the maximum velocities considered. The difference is primarily due to the differences in the critical temperature and in the latent heat of vaporization of Freon-11 and water.

The expendable coolant systems and the expendable portion of combination systems is practically unaffected by flight velocity. Any variation is caused by heat that is picked up from the surroundings by the storage tank, the lines, and the equipment.

C. Flight Altitude

For a given specified flight velocity and equipment exit temperatures, the effect of altitude on the various systems is rather similar for the systems utilizing an expendable coolant. As pointed out previously, the temperatures that can be maintained by means of expendable coolants are determined by the boiling point which in turn is a function of the altitude (figure 89). Within the temperature range of an expendable coolant, the weights will be practically independent of altitude in so far as equipment cooling is concerned. However, unless the storage tank is pressurized, some fluid will be boiled off to bring the fluid temperature to an equilibrium level and to maintain it at that temperature as heat is absorbed from surroundings. The heat absorbed will depend on the conductance of the tank and on the temperature difference between the surroundings and the tank. Thus it is apparent that fluids with a high vapor pressure will require insulation and storage of the coolant under pressure to prevent loss of the coolant by evaporation from the tank, especially at the higher altitudes.

The vapor cycle and the air cycle systems studied exhibit a minimum weight for flight at 40,000 feet altitude. This is due to the temperature schedule (figure 2) and because the heat transfer coefficients applicable to air heat transfer have not dropped too low at this altitude. At higher altitudes, the equivalent weight increases, moderately for vapor cycle cooling systems and for regenerative air cycle cooling systems but quite drastically for simple air cycle cooling systems. The increased weight is due to the reduction in heat transfer at the low atmospheric pressure. At altitudes below 40,000 feet, the weight also increases because of the higher temperatures that prevail. This effect is not considered as significant because normally flight will not be at the high velocities at the lower altitudes.

D. Total Equivalent Weights of Various Systems

A comparison of the total equivalent weight of different systems to be entirely valid requires a rather detailed specification of the equipment cooling requirements and of the aircraft flight schedule. Without such information, an absolute evaluation cannot be made because a system that is outstanding in one respect may be deficient in another respect. Some of the general system characteristics will be discussed as they affect the total equivalent weight for various flight conditions. As pointed out in the general discussion of equipment conditions above and in section IV, a comparison is rather arbitrary and actual evaluation will have to take specific operating requirements into consideration.

At flight speeds up to Mach 1.8, the simpler cooling systems appear to have an advantage. The simple air cycle system imposes a somewhat smaller weight than does the regenerative air cycle system. The Freon-11 and the water vapor cycle systems, assuming a surface condenser, are approximately equal for such conditions and impose a total equivalent weight approximately the same as the air cycle system for equal average equipment temperatures (when the heat transport system and transfer surfaces are included). The vapor cycle systems impose more dead weight and require electrical power (pneumatic or hydraulic could be used). The air cycle systems are relatively light weight but impose a momentum drag and an engine air bleed. Vapor cycle systems using ram air condensers would impose approximately a 20% greater total equivalent weight than the other vapor cycle systems. The increase is primarily due to momentum drag and to higher condenser temperatures. Expendable coolant systems are suitable for certain applications depending on flight time.

The differences in the various systems become more significant at flight speeds from Mach 1.8 to 2.2 at 40,000 feet and at lower flight speeds at higher or lower altitudes as indicated by the operating limit curves, e.g., figures 45 to 48, 76, 77, 85, and 86.

Under flight in this range of velocities, cooling problems have appeared that point to the need for refinements and for more elaborate systems or different fluids. Within this range, the regenerator air cycle system and the simple air cycle are about equal to the regenerative air cycle, having a slight advantage at Mach 2.2. In this range, the Freon-11 vapor cycle system also approaches its limit requiring about 25% greater total equivalent weight than a water vapor cycle system. A cascaded cycle could be used to decrease the total equivalent weight of the Freon-11 system by about 10%. Expendable coolant systems would be applicable, again depending on the flight time required.

Flight at Mach 2.5 imposes conditions for which a regenerative air cycle system (or other modified air cycle system), a water vapor cycle system, or an expendable coolant system is required. This regime is beyond the range of simple air cycle systems and Freon-11 vapor cycle systems. The relative merits of the regenerative air cycle system and the water vapor cycle system depend to a large extent on the particular application. Either system apparently could be designed to meet the cooling requirements. Expendable coolant systems become relatively more attractive as the flight speed increases and would be applicable for somewhat longer flight times at the high velocities than at the lower velocities.

E. Spatial Requirements

In some applications of equipment cooling systems, the system volume may be as significant as weight, power, and drag. The volume and size of some of the components for certain systems are given in the analysis; for other systems volume will depend on design details and is therefore difficult to ascertain with a reasonable degree of accuracy. There are, however, a number of trends that can be pointed out and that may be of considerable significance depending on the application.

Air cycle cooling systems require relatively large air ducts for air flow from the diffuser and engine bleed to the air cycle system and from the system to the equipment. Air-to-air heat exchangers are somewhat larger for a given heat load than are liquid-cooled condensers or evaporators. However, the air cycle turbine and blower is a very compact unit so that the air cycle cooling unit with the necessary heat exchangers requires considerably less space than the vapor cycle compressor, electric motor, and evaporator. The heat transport lines used with the vapor cycle system would be small tubing and therefore require little space.

The vapor cycle systems using a surface condenser or heat exchanger require considerable surface area for heat transfer, typically between one and two square feet per kilowatt. The use of ram air-cooled condensers with a vapor cycle system would of course entail air ducts even larger than are required for air cycle systems. Expendable cooland systems require a tank for fluid storage. The size is of course dependent on the flight duration for which the system is designed.

F. Vulnerability and Dependability

Because of the importance of the equipment cooling system to the operation of the aircraft, the vulnerability and dependability of the system is an important consideration. An evaluation of such factors without verifying statistics can only be relative. The estimated order of increasing vulnerability of the system is (1) ram air systems, (2) simple and regenerative air cycle systems, (3) expendable systems, (4) vapor cycle systems with ram air condenser, and (5) vapor cycle with surface condenser. Vapor cycle systems are somewhat more vulnerable to shell and flak damage than other systems because even a minor leak can render the system inoperative. The surface condenser involves considerable area and therefore puts it at the bottom of the list. The variation in dependability of the various systems is perhaps less than variations in vulnerability but approximately the same order would apply, i.e., with number one being the most dependable system.

G. Centralized Versus Individualized Cooling System Applications

The actual analysis of the various cooling systems was based on systems of 10 kw cooling capacity. However, the values as given are representative in most cases of system characteristics for somewhat smaller systems and for systems several times as large. Smaller systems would in general involve somewhat heavier components on a per kilowatt basis, an increase of say 10 to 15% for systems half as large. Larger units would result in a weight saving on a per kilowatt basis but most systems would have to be made several times as large to secure a 10% weight advantage. An exception to the above general rule is simple ram air systems for which a slight saving in weight may result in some cases with smaller systems because of a possible reduction in total duct weight by using a number of small short ducts.

With the above considerations, the concept of centralized cooling systems versus individualized cooling systems is in most cases reduced to the overall effects of several systems as opposed to the use of a heat transport fluid or the use of air ducts to convey air to different equipment components. A basic consideration is then the complexity and duplication involved in individualized systems. As pointed out in section IV, the transport of heat by means of a heat transport fluid involves a relatively small equivalent weight. The systems considered in this study that utilize a heat transport fluid are rather complex systems with the exception of the expendable coolant system. Even with the latter system, a single storage tank appears preferable. Consequently, all systems using heat transport fluids are considered primarily suitable as centralized systems.

The total equivalent weight will then be lowest and the general system characteristics will be utilized to the best advantage. Likewise, air cycle systems are considered too complex to be suitable for individualized cooling systems.

The distribution lines for the expendable system, such as illustrated in figures 90 and 91, are very light so that this system is also applicable to the centralized concept. These particular systems would be suitable to the individualized applications and would then involve slightly less weight; however, each storage tank would then require individual attention.

Of the systems considered in this study, the ram air cooling system combined with an expendable coolant system as illustrated in figure 94 is considered the most suitable for individualized cooling applications. In this case, the expendable coolant would be stored in a central tank with small tubing to conduct the fluid to the boiler. Individual ram air and fan circulating systems would be used for each component or group of components. The length and size of air ducts could thus be minimized. The total equivalent weight of a number of individual systems would then be less than for a single centralized system of this type.

NOMENCLATURE

Symbol	Definition
A	Area
Br	Breguet
Btu	British thermal unit
b	Thickness
C	Rate of climb
c	Specific heat
D	Drag
Du	Flight duration
d	Diameter
e	Effectiveness
F	Dimensionless factor
Fa	Radiation shape factor
Fe	Radiation emissivity factor
f	Equivalent weight factor
g	Acceleration of gravity
H	Altitude
HP	Horsepower
h	Heat transfer coefficient or enthalpy
I	Input
j	Heat exchanger constant
k	Conductivity
kw	Cooling load
L	Latent heat or lift
ℓ	Length
M	Mach number
N	Number of passes - heat exchanger
Nu	Nusselt number
n	Compression exponent
P	Power
Pr	Prandtl number
p	Pressure
Q	Heat flow or heat quantity
R	Gas constant
Re	Reynolds number
Rg	Range
r	Pressure ratio
SFC	Specific fuel consumption
s	Distance along surface
T	Temperature or thrust

Symbol	Definition
t	Time
U	Overall heat transfer coefficient
V	Velocity or volume
v	Specific volume
W	Weight
w	Fluid flow rate
X	Fuel to gross weight ratio (W_f/W_g)
x	Linear distance
z	Leakage flow ratio (w/w_a)

Greek Symbols

γ	Ratio of specific heats (c_p/c_V) or specific weight
Δ	Increment
δ	Normalized pressure ($p/2116$, where p is lb/ft^2)
ζ	$(wc_p)_a/(wc_p)_b$
η	Efficiency
θ	Normalized temperature ($T/519$, where T is in °R)
λ	Temperature ratio across air turbine (T_{E_3}/T_{t1})
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Density
σ	Stephan-Boltzmann constant
ϕ	Heat exchanger pressure drop parameter
Ω	Heat exchanger heat transfer parameter ($UA/3600\ wc_p$)

Subscripts

A	Average
a	Ambient
aft	Afterburner
B	Blower
b	Bleed
bo	Boiler
C	Compressor
D	Drag
d	Direct
Diff	Diffuser
dt	Duct
E	Equipment
e	Exit
eq	Equivalent
ev-a	Evaluation average
ex	Exchanger
exp	Expendable

Subscript	Definition
f	Fuel
G	Gross
g	Gas
I	Intermediate
i	Inlet or inside
K	Condenser
ℓ	Liquid
M	Motor and generator
max	Maximum
mom	Momentum
N	Net
n	Nozzle
o	Outside
P	Power
p	Pressure
R	Regenerative
r	Recovery
S	Surface
s	Storage
sub	Subcritical
sup	Supercritical
T	Total
t	Turbine
th	Thrust
V	Evaporator
v	Volumetric
W	Wall
x	Linear distance

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APPENDIX I

HEAT EXCHANGER ANALYSIS

The heat exchanger characteristics utilized in the main body of the report are based upon the detailed analysis given herein. Three types of heat exchangers are considered: air-to-air, liquid-to-air, and the evaporative (or boiler) heat exchanger in which air is cooled by boiling of an expendable coolant. The main purpose of the analysis is to establish the required size and weight of these exchangers for specified design values of cooling effectiveness, flow rates of cooled and coolant fluids, and air side pressure losses. Two distinct phases of the heat exchanger analysis can be differentiated: (1) determination of the required heat transfer conductance of the exchanger in terms of its cooling effectiveness, its physical arrangement and the flow rates of cooled and coolant fluids; and (2) determination of the size and weight of the exchanger from characteristics of the core, in terms of permissible pressure losses and the heat transfer conductance found under 1.

1. <u>Required heat transfer conductance.</u> The cooling effectiveness of a heat exchanger transferring heat from fluid 1 to fluid 2 is defined by the relationship

$$e_1 = \frac{T_{1i} - T_{1o}}{T_{1i} - T_{2i}} \tag{171}$$

For a given heat exchanger arrangement, the effectiveness e_1 depends upon two parameters: a conductance parameter Ω and a flow parameter ζ. For the counterflow arrangement shown in figure 56, the effectiveness can be expressed as an explicit function of Ω and ζ:

$$e_1 = \frac{\zeta\left(1 - e^{(\zeta - 1)\Omega}\right)}{1 - \zeta e^{(\zeta - 1)\Omega}} \tag{172}$$

(assuming that the overall conductance U is uniform over the heat transfer surface). The counterflow arrangement enables the highest possible effectiveness for given values of Ω and ζ; however, this configuration is not often practicable for air heat exchangers (especially for high effectiveness) because of headering difficulties. Single or multiple pass crossflow arrangements (figure 56) permit a close approach to the performance of the counterflow exchanger and are particularly well suited to air-to-air and liquid-to-air heat exchangers. A rigorous analytical solution for the crossflow exchanger has been given by Nusselt (reference 16). Multi-pass crossflow configurations are not readily susceptible to analysis unless simplifying assumptions are made. However, a number of crossflow cases have been analyzed numerically by Stevens and Wolfe (reference 21) with the aid of digital computing equipment. These data have been used herein for the analysis of the regenerative heat exchanger, which is discussed subsequently. However, most of the multi-pass crossflow heat exchanger analysis described herein was based upon a simplified analysis wherein it was assumed that the cooled fluid flow (see figure 56) is mixed, while the coolant remains unmixed throughout. The approximate analysis resulted in the following formulae:

<u>Double-pass counter-crossflow</u>

$$e_1 = 1 - \frac{e^{-A\zeta}}{e^{A\zeta} - A \sinh A\zeta} \qquad (173)$$

where $A = 1 - e^{\frac{-\Omega}{2\zeta}}$

<u>Triple-pass counter-crossflow</u>

$$e_1 = \frac{Q - 1}{Q} \qquad (174)$$

where $Q = e^{3A'\zeta} - A'^2 \zeta \left(1 - \frac{A'}{4}\right) e^{A'\zeta} - A' \left(2 - \frac{A'}{4}\right) e^{2A'\zeta} \sinh A'\zeta$

and $A' = 1 - e^{\frac{-\Omega}{3\zeta}}$

These expressions are in agreement with equations given in reference 21. Figure 57 shows the relation between Ω and ζ as determined from equations (2), (3), and (4) for the counterflow, double-pass, and triple-pass cases and effectiveness of 0.8 and 0.9. It is evident that the triple-pass counter-crossflow arrangement closely approaches the counterflow arrangement in theoretical efficiency for $\zeta = 1.5$ or greater. It can be proved that the assumption of mixed conditions in the cooled fluid is always conservative (i.e., leads to a larger required Ω for given e_1 and ζ) as compared with the mixed case.

In the case of the evaporative heat exchanger, the effectiveness is simply

$$e_1 = 1 - e^{-\Omega} \tag{175}$$

2. Heat exchanger core characteristics. Air side heat transfer conductances and pressure loss data for many types of core construction are given in the literature. In general, these data can be expressed, with sufficient accuracy for analytical purposes, in the form (see figure 59 for definition of dimensions):

Heat transfer conductances (single pass)

$$(hA)_1 = a_1 \left(\frac{w_1}{\ell_c \ell_n}\right)^{p_1} \ell_c \ell_s \ell_n \tag{176}$$

$$(hA)_2 = a_2 \left(\frac{w_2}{\ell_s \ell_n}\right)^{p_2} \ell_c \ell_s \ell_n \tag{176a}$$

Core pressure losses (single pass)

$$\overline{\rho}_1 \Delta p_1 = b_1 \left(\frac{w_1}{\ell_c \ell_n}\right)^{q_1} \ell_s \tag{177}$$

$$\overline{\rho}_2 \Delta p_2 = b_2 \left(\frac{w_2}{\ell_s \ell_n}\right)^{q_2} \ell_c \tag{177a}$$

where a_1, a_2, b_1, b_2 and the exponents p_1, p_2, q_1, q_2 are constants for a particular core under turbulent flow conditions. In cases where very accurate core data are available, inlet and exit pressure losses may be taken into account, however, in the present instance, these losses are accounted for approximately by suitable increases in b_1 and b_2 to permit equations (177) and (177a) to be used for overall core pressure losses. When the core pressure losses are a small fraction of the inlet absolute pressure (as is almost always the case), equations (177) and (177a) can be written in the following, more convenient, form:

$$\phi_1 = \left[\frac{\delta_i^{\,2}}{\theta}\left(\frac{\Delta p}{p_i}\right)\right]_1 = 0.0062 b_1 \left(\frac{w_1}{\ell_c \ell_n}\right)^{q_1} \ell_s \tag{178}$$

$$\phi_2 = \left[\frac{\delta_i^{\,2}}{\theta}\left(\frac{\Delta p}{p_i}\right)\right]_2 = 0.0062 b_2 \left(\frac{w_2}{\ell_s \ell_n}\right)^{q_2} \ell_c \tag{178a}$$

The core data applied in analyzing heat exchanges in this study are summarized in table 5.

3. <u>Determination of the size and weight of a heat exchanger for specified performance.</u> When the effectiveness e_1, flow rates w_1 and w_2, and allowable pressure losses are known, the required size and weight of a crossflow exchanger is found in the following way: (1) As described under item 1, determine $\Omega = UA/(3600wc_p)_1$ and then UA for the specified values of e_1, w_1 and w_2. (2) For a given core, express the overall conductance UA in the general form:

$$UA = (3600wc_p)_1\,\Omega = \frac{a_1\left(\dfrac{w_1}{\ell_c \ell_n}\right)^{p_1} N(\ell_c \ell_s \ell_n)}{1 + \dfrac{a_1}{a_2}\left(\dfrac{w_1}{\ell_c \ell_n}\right)^{p_1}\left(\dfrac{w_2}{\ell_s \ell_n}\right)^{-p_2}} \qquad (179)$$

(3) By using equations (178) and (178a) together with equation (179), find the volume of the heat exchanger $N(\ell_s \ell_c \ell_n)$ in terms of e_1, ζ, w_1, Φ_1, and Φ_2; the weight is then found from

$$W_{ex} = \gamma N(\ell_c \ell_s \ell_n) \qquad (180)$$

where γ is found from table 5. Application of this procedure is discussed below for each of the heat exchanger types considered herein.

(a) Ram heat exchanger (air cycle cooling systems). A stainless steel tubular core is assumed because of the high bleed air temperature. Using data from table 5, equation (179) can be rearranged to give

$$N\ell_s = 2.38\,\Omega c_p\left[x^{0.2} + 0.005\left(\frac{\ell_s}{\zeta \ell_c}\right)^{0.6} x^{0.4}\right] \qquad (181)$$

where $x = \dfrac{w_b}{\ell_c \ell_n}$

From equations (178) and (178a)

$$_b = 9.35X^{1.8}\,\ell_s\,N\,10^{-3} \qquad (182)$$

TABLE 5 HEAT EXCHANGER CORE CHARACTERISTICS

Description	Application	a_1	a_2	b_1	b_2	p_1	p_2	q_1	q_2	y*
Tubular: 5/32 O.D. equilateral pitch with transverse spacing of 0.3 inches (Reference 19)	Ram Air Heat Exchanger (air-to-air)	1513	2500	1.5	1.5	0.6	0.6	1.8	1.8	47.8
Serrated plate-and-fin extended surface (Reference 20)	Regenerative Heat Exchanger (air-to-air) Evaporative Heat Exchanger (air-to-liquid)	5500	5500	3.0	3.0	0.635	0.635	1.635	1.635	31.3
Finned tubular extended surface (Reference 22)	Intermediate Heat Exchanger (liquid-to-air)		3600		1.0		0.517		1.762	20.0

*With allowance for housing, diffusers, etc.

$$\phi_a = 9.35 \left(\frac{x\zeta \ell_c}{\ell_s}\right)^{2.8} \ell_c N\, 10^{-3} \qquad (183)$$

Solving equation (182) for $N\ell_s$ and substituting in equation (181)

$$\phi_b = 0.00535\,\Omega\left[x^{2.0} + 0.605\left(\frac{\ell_s}{\zeta \ell_c}\right)^{0.6} x^{2.2}\right] \qquad (184)$$

Using equations (180) and (182), the weight of the exchanger is

$$W_{ex} = 47.8\, N\,(\ell_c \ell_s \ell_n)$$

$$= 47.8\,(N\ell_s)\,\frac{w_b}{x} = 5120\,\frac{\phi_b w_a}{x^{2.8}} \qquad (185)$$

The cooling flow pressure drop is (from equations 182 and 183)

$$\phi_a = \frac{\phi_b}{\zeta}\left(\frac{\zeta \ell_c}{\ell_s}\right)^{2.8} \qquad (186)$$

Using these equations, the ram heat exchanger weight was computed for the following assumed conditions: $N = 3$ (counter-crossflow arrangement) $\ell_s/\ell_c = 3$, $\zeta = 1.5$, e_b ranging from 0.5 to 0.8. The results are plotted in figure 61 with e_b and ϕ_a as independent variables.

(b) Regenerative heat exchanger (regenerative air cycle cooling system). The regenerative exchanger analysis parallels that for the ram exchanger given above. From equations (178) and (178a)

$$\phi_t = 0.00293\,\Omega x^{2.0}\left[1 + \left(\frac{\ell_s}{\ell_c}\right)^{0.635}\right] \qquad (187)$$

(data from table 5).

Making use of equations (178) and (187),

$$\ell_s N = 0.157\,\Omega x^{0.365}\left[1 + \left(\frac{\ell_s}{\ell_c}\right)^{0.635}\right] \qquad (188)$$

The weight of the exchanger is then

$$W_{ex} = 31.3\,(N\,\ell_s)\,(\ell_c\,\ell_n)$$

$$= 49.2\,\Omega\,x^{-0.635}\,w_1\left[1 + \left(\frac{\ell_s}{\ell_c}\right)^{0.635}\right] \qquad (189)$$

Solving equation (187) for x and substituting in equation (189)

$$\frac{W_{ex}}{w_1} = 67\,\frac{\Omega^{1.317}}{\phi_1^{\,0.317}}\left[1 + \frac{\ell_s}{\ell_c}^{\,0.635}\right]^{1.317} \qquad (190)$$

The cooling airflow pressure loss parameter is, from equations (178) and (178a)

$$\phi_2 = \phi_1\,(\ell_c/\ell_s)^{2.635} \qquad (191)$$

Choosing $\ell_s/\ell_c = 20$ to keep the cooling flow pressure losses reasonably low, the regenerative exchanger weight was computed from equation (190) for $e_1 = 0.75$, 0.80, and 0.85; the results are plotted versus ϕ_1 in figure 62.

(c) Intermediate heat exchanger (regenerative air cycle cooling system). Assuming that the transfer fluid side conductance is very large compared with the air side conductance, the weight of the intermediate exchanger is found to be approximately

$$\frac{W_{ex}}{(wc_p)_\ell} = 3.37\,\Omega^{1.232}\,(\zeta\,\phi_a)^{-0.232} \qquad (192)$$

The derivation of this equation is similar to the preceding analysis and was based upon core data from table 5. The assumed arrangement of the intermediate exchanger is illustrated in figure 32. The relationships expressed by equation (192) is shown in figure 79 for $e_\ell = 0.8$ and $\zeta = 1.0$ and 1.5. The conductance factor Ω was computed from equation (172) on the assumption that four or more transverse passes are taken by the transfer fluid.

(2) Evaporative (boiler) heat exchanger (ram air-expendable coolant cooling system). By a straightforward analysis similar to the foregoing, the weight of the evaporative exchanger was found to be expressible by the equation

$$\frac{W_{ex}}{w_b} = 1.42 \frac{\Omega^{1.317}}{\phi_{bo}^{0.317}} \qquad (19$$

where Ω is given by equation (175). Equation (193) is plotted in figure 63 for $e_b = 0.85$, 0.90, and 0.95. A sketch of the exchanger is shown in figure 60.

APPENDIX II

EVAPORATOR AND CONDENSER ANALYSIS

In the analysis of vapor cycle cooling systems, it was pointed out that the evaporator temperature has a very significant effect on the system total equivalent weight. A typical evaporator evaluation procedure is outlined below indicating the assumptions that must be made during the analysis and the general procedure that was used to secure an engineering optimization of the overall cooling system. Similar analyses were made to determine condenser and surface heat exchanger weights and optimum configurations. The calculations and optimization were performed by tabular and graphical means.

In the system analysis, the cooling load (kw), the equipment exit temperature (T_{Ee}), and transport line lengths were taken as independent variables. The refrigerant and transfer fluid properties were determined at the applicable temperatures. The evaporator effectiveness (e_V), the evaporating temperature (T_V), the condensing temperature (T_K), and the fluid flow velocities were taken as design variables and were subjected to a graphical optimization procedure.

The effectiveness of an evaporator is defined as

$$e_V = \frac{T_{Ee} - T_{Ei}}{T_{Ee} - T_V} \qquad (194)$$

This effectiveness can also be given in terms of the overall heat transfer parameter (UA) and the thermal flow rate ($3600wc_p$) by the equation

$$e_V = \frac{e^z - 1}{e^z} \qquad (195)$$

where $z = UA/3600wc_p$

Equation (195) is plotted versus $UA/3600wc_p$ in figure 120.

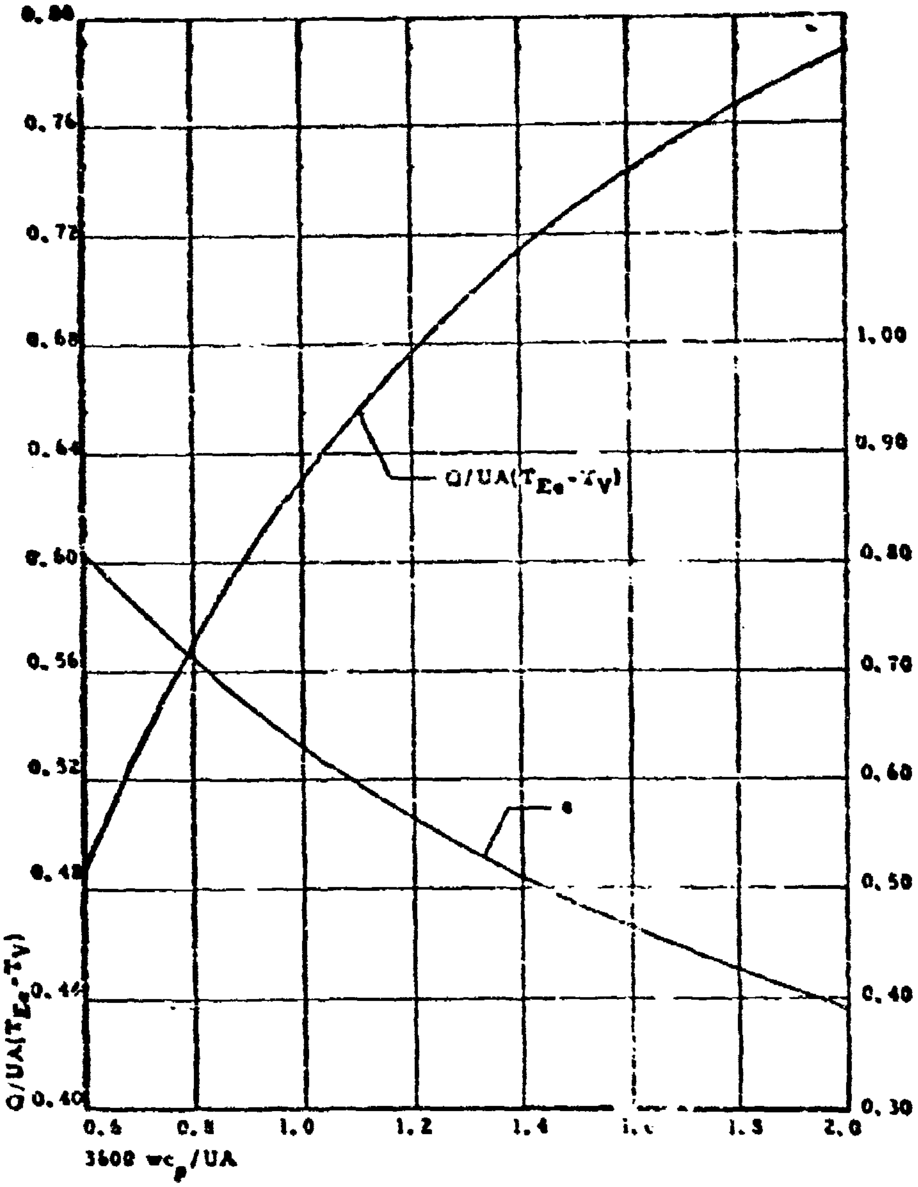

FIGURE 120 EVAPORATOR EFFECTIVENESS AND HEAT TRANSFER PARAMETER

By assuming an ϵ_V, the required UA/wc_p can be determined from figure 120. (Various assumptions are made to obtain an optimum value.)

The required value of wc_p can be determined by considering the heat that must be absorbed by the transport fluid

$$wc_p = \frac{Q}{3600\, \epsilon_V \,(T_{Ee} - T_V)} \qquad (196)$$

The required value of UA can be determined by means of equation (196) and figure 120.

Assuming an evaporator tube diameter, the required number of parallel tubes and transport fluid flow velocity can be determined from the equation

$$nV = \frac{4w}{\pi d^2 \gamma} \qquad (197)$$

Assuming a number of parallel tubes and a flow velocity consistent with equation (197), the heat transport fluid heat transfer coefficient (h_i) can be determined from the equation (assuming turbulent flow Re > 2300)

$$h_i = 0.023 \frac{k}{d} (Re)^{0.8} (Pr)^{0.4} \qquad (198)$$

Equation (198) is plotted versus flow velocity in figure 121 assuming an ethylene glycol solution. An ordinate scale is included in figure 121 with a heat transfer parameter that is independent of fluid properties.

Values of evaporative heat transfer coefficients are given in the literature (references 1, 9, and 15). The effective heat transfer parameter UA is

$$UA = \frac{h_1 A_1 h_2 A_2}{h_1 A_1 + h_2 A_2} \qquad (199)$$

Values of $UA/h_2 A_2$ are plotted versus $h_2 A_2/h_1 A_1$ in figure 122.

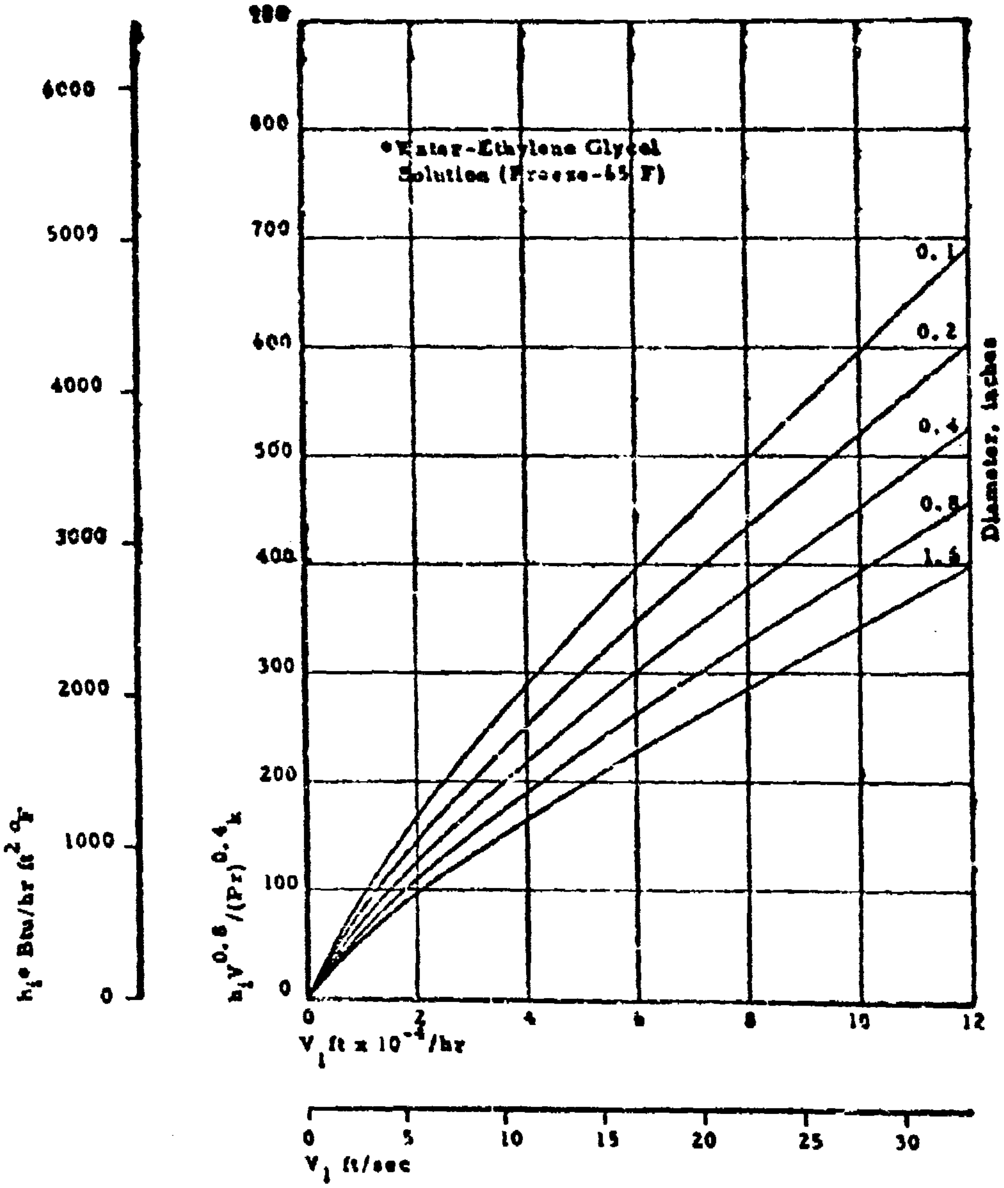

FIGURE 121 HEAT TRANSFER COEFFICIENT FOR LIQUID FLOWING IN TUBES

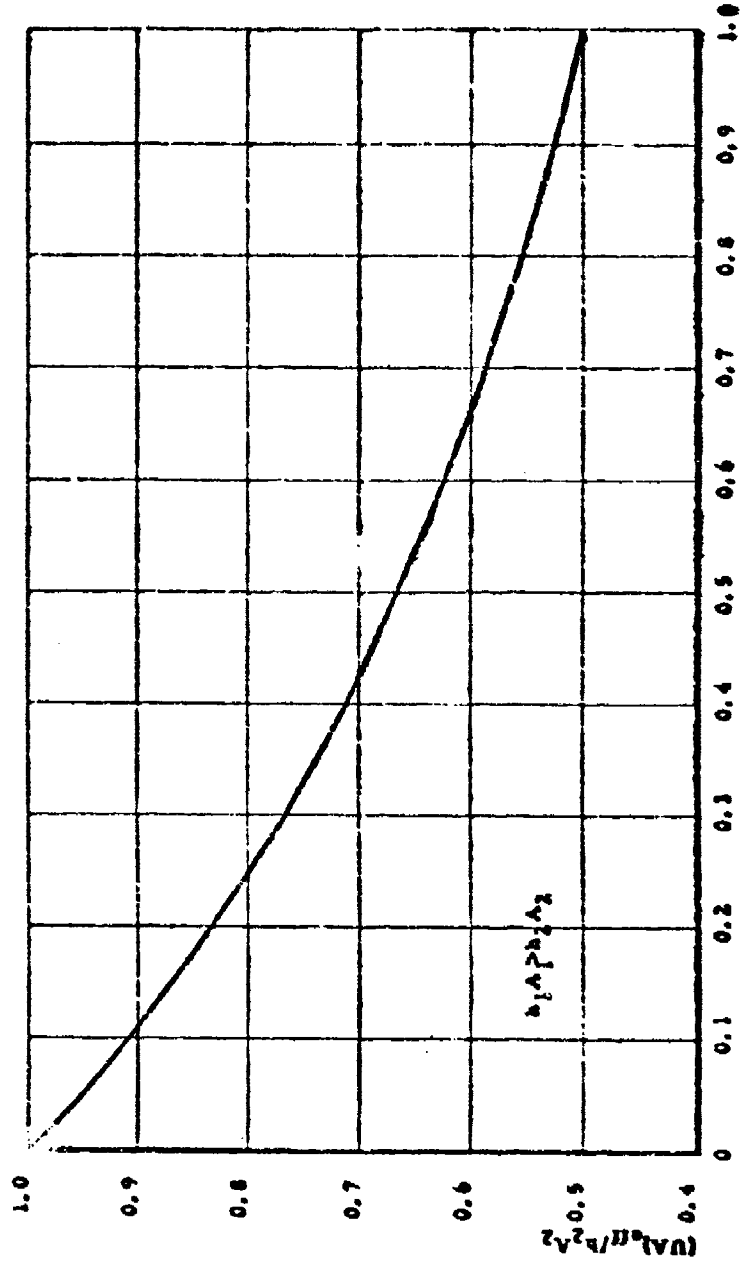

FIGURE 122 EFFECTIVE HEAT TRANSFER PARAMETER

The required tubing length is determined by means of equations (196), (199), and figure 120. The evaporator weight is determined on the basis of the required tubing length and the weight per unit length including transport fluid with a factor allowed for the shell, headers, and the refrigerant.

The power required to circulate the heat transport fluid through the evaporator is determined from the equation

$$HP_f = C\, f_f\, V^3\, d\, \ell \tag{200}$$

The constant C depends on the units used. The fluid function factor f_f can be found from curves in reference 25, page 250, or in books on fluid dynamics. Equation (200) is also used to determine power required to circulate the heat transport fluid. The size of heat transport lines can be optimized by using a weight equivalent for the power and then minimizing the sum of the equivalent and actual weights of the transport system.

The condenser is analyzed in a very similar manner except that for a surface condenser, the effectiveness is not defined because the recovery temperature is assumed constant (equivalent to assuming an infinite heat-sink) and condensation is assumed to be a thermal isothermal process. The condenser is therefore analyzed as outlined in section V.

The condenser and surface heat exchanger portion of the vapor cycle systems using a liquid cooled condenser and surface heat exchanger (figure 35) can be analyzed in a manner similar to that outlined for the evaporator above with modifications to account for the surface condenser. The liquid-cooled condenser is treated like the evaporator with the heat transport fluid removing the heat of vaporization and using the applicable condensation heat transfer coefficient. For the higher temperatures, a heat transport fluid such as Dowtherm A is used for this part of the system. The effectiveness curve of figure 120 applies to the condenser and surface heat exchanger. Figures 121 and 122 can also be used in the condenser and surface heat exchanger analysis. The heat transfer parameter for this case with the separate condenser and heat exchanger is given by the equation

$$\frac{Q}{(UA)_1 (T_K - T_r)} \quad \frac{}{(UA)_1}\left[\left(\frac{e^{z_1}}{e^{z_1} - 1}\right) + \left(\frac{e^{z_2}}{e^{z_2} - 1}\right) - 1\right]^{-1} \tag{201}$$

where $z_1 = (UA)_1/3600wc_p$

$z_2 = (UA)_2/3600wc_p$

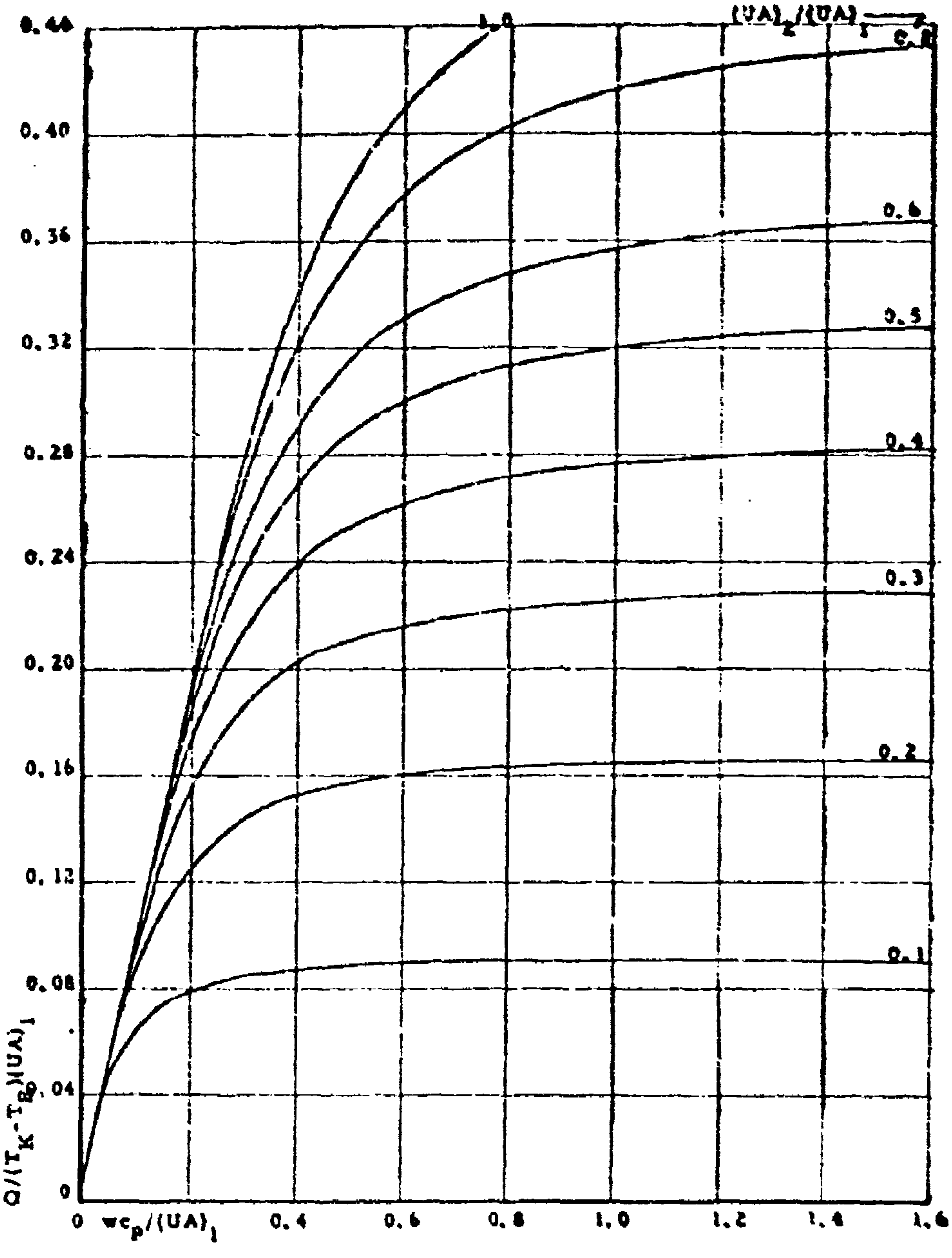

FIGURE 123 HEAT TRANSFER PARAMETER FOR LIQUID COOLED CONDENSER AND SURFACE HEAT EXCHANGER

The heat transfer parameter as given by equation (201) is plotted versus $wc_p/(UA)_1$ in figure 123 for various values of $(UA)_2/(UA)_1$.

The weight of the compressor and the power supply system, and the power requirement, for the assumed evaporating and condensing temperatures are determined as discussed in section V.

A tabulation procedure is used to determine the effect of varying the design variables. Translation factors are used to determine weight equivalents for the power requirements and tabular and graphical methods are applied to determine the minimum total equivalent weight.